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NATO Committee on the Challenges of Modern Society

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TECHNICAL REPORT
OF THE
CONFERENCE ON LOW POLLUTION
POWER SYSTEMS DEVELOPMENT

NATO Committee on the Challenges of
Modern Society

Eindhoven, The Netherlands
February 23, 24, 25, 1971

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Introduction

The NATO/CCMS Conference on Low Pollution Power Systems Development was held in Eindhoven, The Netherlands, on February 23-25, 1971. The Government of The Netherlands and Philips Company co-hosted the Conference.

The total attendance for the meeting was just under one hundred government and industrial representatives from twelve countries, as well as non-NATO international organizations. The following countries were represented: The Netherlands, Belgium, Canada, Denmark, France, Germany, Great Britain, Italy, Turkey, Sweden, Japan, and the United States. The Organization for Economic Cooperation and Development (OECD) and the European Economic Community had observers in attendance.

The Conference was divided into two sessions with the first day reserved for government representative presentations and discussions regarding country programs and policies relative to the automotive pollution problem. The second and third days were devoted to technical presentations on research and development projects by both government and industry attendees.

The "Summary of Proceedings of the Conference on Low Pollution Power Systems Development"* was published by the U. S. Environmental Protection Agency (EPA) in March 1971. The Summary of Proceedings document gives an overview of the Conference and includes brief summaries of each presentation made at the Conference by government and industry representatives.

*This document is available from the Environmental Protection Agency.

This document includes the texts and figures of the technical presentations that dealt with research efforts in the low pollution power systems field. Additionally, the one technical paper dealing with the rationale for the U. S. standards was also included because of the extensive amount of information on the relationship between air quality and automobile emissions contained in the paper. The U. S. companies who participated in the conference are contractors to the Environmental Protection Agency for low pollution power systems development. Other discussion papers, which were general in nature relating to governmental concern with the automobile pollution problems, were not included; however, copies of these papers may be obtained from the participants. The summaries are included in the Summary of Proceedings document.

Chapter II

FEDERAL MOTOR VEHICLE EMISSION GOALS FOR CO, HC, AND NO_X BASED ON DESIRED AIR QUALITY LEVELS*

by

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Federal Regulations

Background

The Federal motor vehicle air pollution control program has its origin in the Clean Air Act, as amended in 1965.[1,2] This Act borrowed heavily from the motor vehicle air pollution control program already in effect in California[3,4] and reflected testimony from the automotive industry that similar controls were feasible for application nationwide. The automotive industry favored national standards to forestall a proliferation of regulatory legislation at the state level. National standards, applicable to the 1968 model year, were promulgated on March 31, 1966.[5] More stringent and more equitable national standards, applicable to the 1970 model year, were promulgated on June 4, 1968.[6] In addition, on February 10, 1970, Secretary Finch published an advance notice of proposed rulemaking indicating the Department's intent to adopt more stringent standards for the 1973 and 1975 model years.[7]

Air Quality Criteria

Under the 1967 amendment to the Clean Air Act,[8] the Department of Health, Education, and Welfare is required to publish air quality criteria Documents. With criteria for photochemical oxidants,[9] carbon monoxide,[10] hydrocarbons,[11] and nitrogen oxides either published or underway it now becomes possible to consider future motor vehicle emission standards in terms of desired air quality consistent with information in the pertinent air quality criteria documents. The resumes of these documents describe what in the Secretary's judgment are the concentrations and exposure time which cause or contribute to, or are likely to cause or contribute to, air pollution which endangers human health or welfare.

Desired Air Quality Goals

Based on the criteria resumes and observed aerometric relationships, Table I has been constructed to show the maximum values which would seem to be consistent with health-related criteria. Minimum safety margin considerations have been incorporated into these levels.

Calculating Motor Vehicle Emission Goals

Using the information in Table I, simple roll-back techniques, like those used by California, could be used to calculate needed emission reductions. These techniques, however, involve a number of assumptions which may not be entirely valid. For example, inherent in the techniques employed is the assumption that the increase in atmospheric concentrations of primary pollutants will be directly proportional to the growth in emissions of the contaminant. Also, the roll-back techniques employed by California assumed a linear correlation to exist between the severity of the manifestation, specifically oxidant index, and reactive hydrocarbon emissions to the atmosphere. The complexities of the photochemical alterations involving reactive hydrocarbons in the atmosphere are too great to permit such an assumption, a priori, because at a minimum the role of the oxides of nitrogen must be taken into consideration.

Clearly what is needed is to fully relate all these effects to emissions and to predict future events in growth and required control

Table I. Desired Air Quality Goals for Health Protection[9-11]

Contaminant	Concentration, g/m^3	Average, hr
Carbon Monoxide	$\leq$ 10,000 (9 ppm)	8
Photochemical Oxidants	$\leq$ 125 (0.06 ppm)	1
Nitrogen Dioxide[a]	$\leq$ 190 (0.10 ppm)	1

[a] Preliminary estimated value pending review of technical report on direct health effects of nitrogen dioxide and publication of Air Quality Criteria Document.

using a comprehensive simulation model complete with modules reflecting all input variables, meteorological variables including air transport parameters, and most importantly-where oxidants, eye irritants, and aerosols are concerned-the chemical kinetics describing the multiplicity of the reactions which occur. Factual data on existing air quality are also required in order to validate and adjust the model as necessary. Several studies are currently investigating the generation of these simulation models.[12,13] Unfortunately, none of these models is yet ready for general application.

Relating Air Quality Goals for Nitrogen Oxides and Nonmethane Hydrocarbons to Oxidant Goals

In the absence of a validated model which incorporates atmospheric chemical mechanisms, a more restricted diffusion model may be employed in which empirical relationships between primary atmospheric pollutants and secondary reaction products are known. Such relationships have been established between photochemical oxidants and nonmethane hydrocarbons.[11] Analogous relationships have also been developed for the oxides of nitrogen separately as well as in combination with nonmethane hydrocarbons. One important premise, based on experiences reflecting the cause and effects associations observed to relate Los Angeles with Pasadena, is that hydrocarbon emissions between the hours of 6 and 9 a.m. result in peak oxidant concentrations 2 to 4 hours later.[11] This is particularly true on the days of greatest interest-days which because of favorable meteorology favor maximum production of oxidant.

Figure 1 includes an analysis of 3 years of data which shows that on those several days a year when meteorological conditions were most

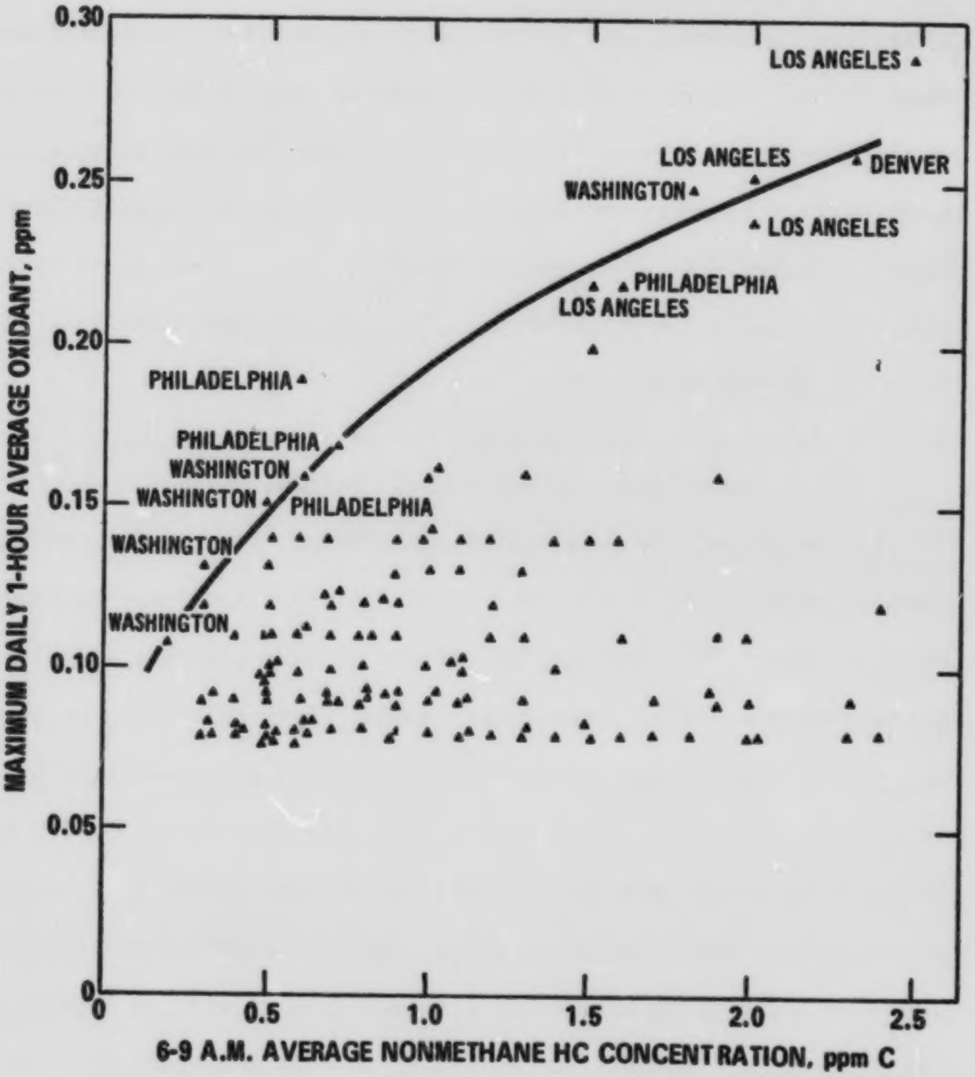

Figure 1. Maximum daily oxidant as a function of early morning nonmethane hydrocarbons; 1966-1968 CAMP stations; May through October 1967 for Los Angeles.11

hydrocarbon concentrations of 200 μg/m^3 (0.3 ppm C) for the 3-hour period from 6 to 9 a.m. might produce an average 1-hour photochemical oxidant peak concentration of up to 200 μg/m^3 (0.10 ppm) 2 to 4 hours later.[11] The hydrocarbon measurements were confined to 200 μg/m^3, or above, because of instrumentation limitations. However, if the functional relationships between the hydrocarbon and photochemical oxidant measurements were extended to include the levels presented in Table III as the highest value consistent with health-related criteria, the corresponding hydrocarbon concentration would be approximately 125 μg/m^3 (0.10 ppm C).

Figure 2 presents similar data relating 6 to 9 a.m. average nitrogen oxides concentrations to maximum daily 1-hour average oxidant concentrations appearing 2 to 4 hours later. The envelope enclosing the data presumably reflects those several days a year when meteorological conditions were most conducive to the formation of photochemical oxidants. On such occasions, if the functional relationships between the oxides of nitrogen and photochemical oxidant measurements were extended to include the level of oxidants presented in Table 1, the corresponding nitrogen oxides concentrations would be approximately 49 μg/m^3 as NO_2 (0.026 ppm).

Figure 3 is a three-dimensional presentation of the same data shown in Figures 1 and 2.[11] The isopleths traced through the different oxidant levels reflect the fact that the ratio of nonmethane hydrocarbon to nitrogen oxides is important, along with the absolute level of the separate independent variables, in determining the maximum level of oxidant produced on days of favorable meteorology. Presentation of the data in this form is useful for qualitatively considering the trade-offs between unilateral and joint control of hydrocarbons

Table III. Relationship of Air Quality Goals to Motor Vehicle Emission Goals

Emission	Type of Effect	Health-Related Air Quality Goals	Health-Related Averaging Time, hr	Emission Goals from all Vehicles to Achieve Desired Air Quality, g/mi	Control Requirement for All Vehicles, %
Carbon Monoxide	Direct	10 mg/m^3 (9 ppm)	8	6.16	92.5
Nitrogen Oxides[a] (NO_2)	Direct (NO_2)	190 $\mu g/m^3$ (0.1 ppm) (NO_2)	1	0.38	93.6
Hydrocarbons	Indirect	125 $\mu g/m^3$ of oxidant (0.06 ppm)	1	0.15	99.0

[a] The NO_x emission goal is identical to the NO_2 emission goal since in Los Angeles, the city at maximum risk, all NO is converted to NO_2.

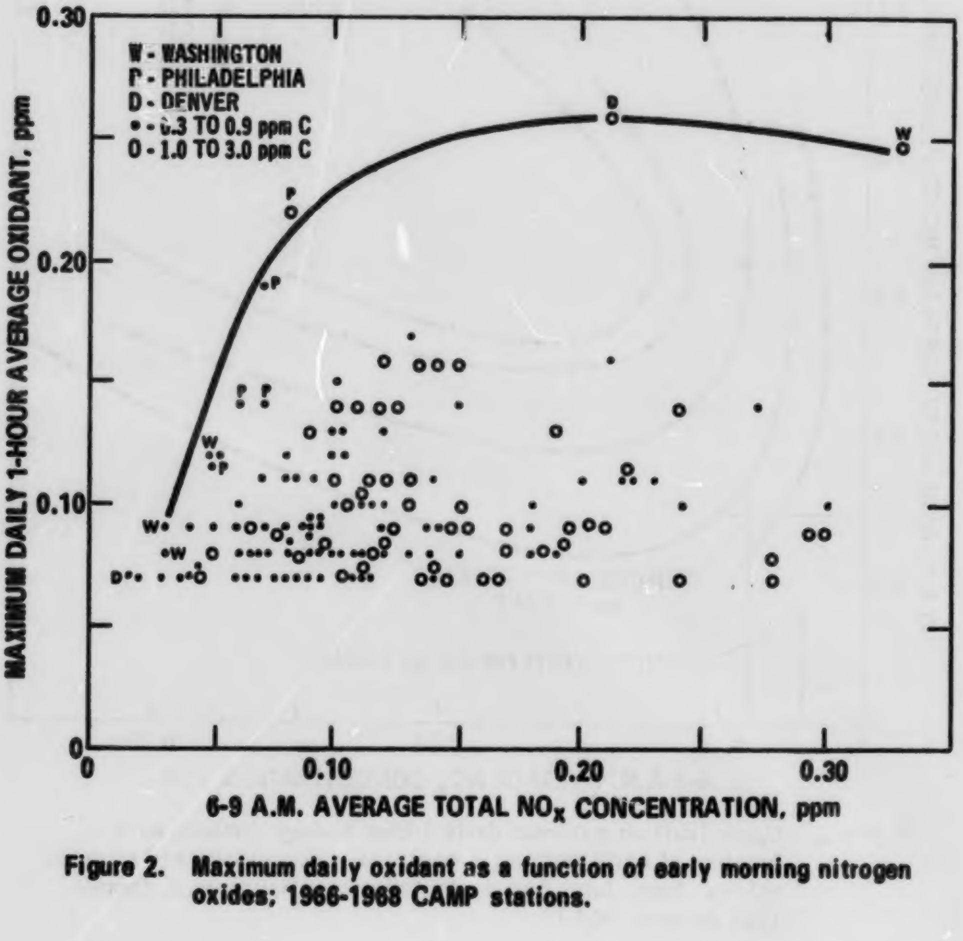

Figure 2. Maximum daily oxidant as a function of early morning nitrogen oxides; 1966-1968 CAMP stations.

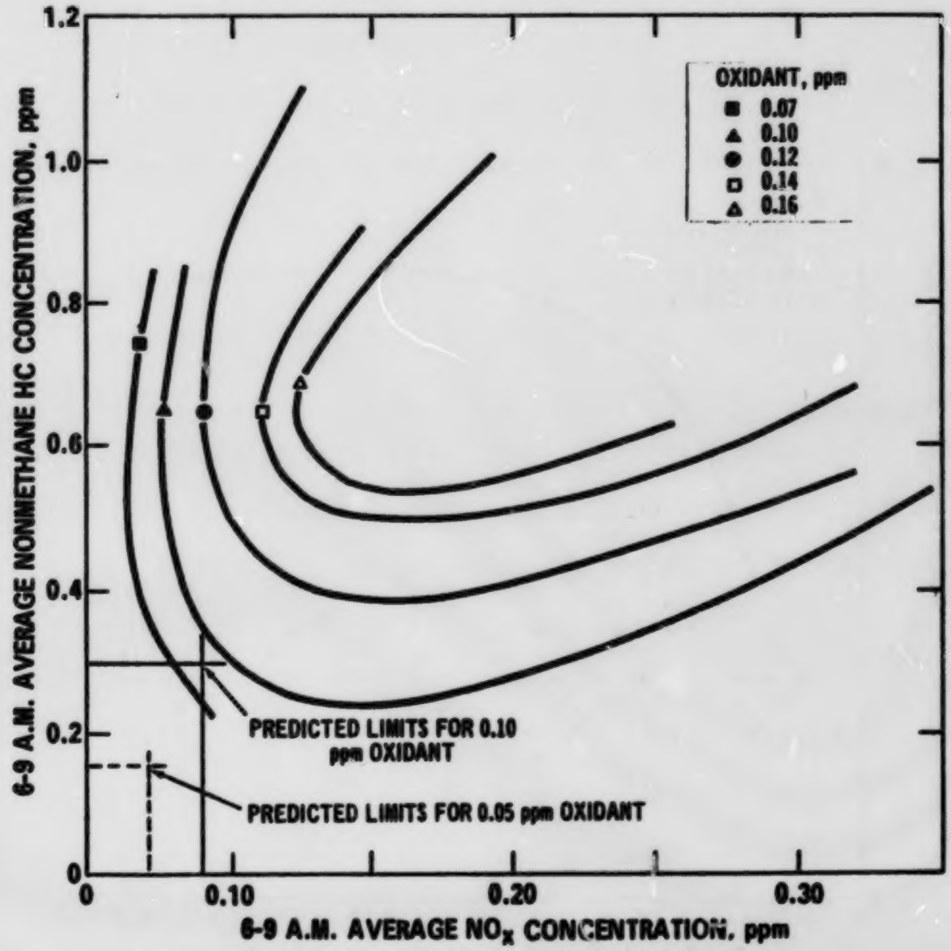

Figure 3. Upper limit on maximum daily 1-hour average oxidant as a function of early morning nonmethane hydrocarbons and nitrogen oxides; June, July, August; Philadelphia, Washington, Denver; 1966 through 1968.[11]

and oxides of nitrogen. It must be emphasized that Figure 3 at this time is based on relatively few points. Thus the curves can be better defined as further data become available. In view of this, the drawing of quantitative conclusions from this graph is not wholly justified.

Qualitative examination of Figure 3 shows that, depending on the specific existing concentrations of oxidants, nitrogen oxides, and nonmethane hydrocarbons, the optimum control strategy for the most rapid reduction of oxidants may be unilateral control of hydrocarbons, unilateral control of nitrogen oxides, or joint control of both nitrogen oxides and hydrocarbons. It must be pointed out, however, that all possible oxidant control strategy options are not available to us. Recall that the postulated health-related desired air quality for NO_2 is 190 $\mu g/m^3$ (0.10 ppm) for a 1-hour average (Table I). Thus no oxidant control strategy is acceptable which will not simultaneously lower the NO_2 values to the desired air quality. Furthermore, our estimate of available nitrogen oxides control technology is that there is at this time no proven or postulated device capable of lowering nitrogen oxides emissions from internal combustion engine mobile sources sufficiently to achieve air quality concentrations of $\leqq$0.026 ppm, which we have shown would be required to control oxidant air quality concentrations to 125 $\mu g/m^3$ (0.06 ppm) by unilateral nitrogen oxides control.

We conclude that the most rational control strategy at this time is to control nitrogen oxides to achieve the desired health-related air quality for NO_2 and to rely on control of nonmethane hydrocarbons to achieve the desired oxidant air quality goal. Subsequent calculations will be based on this approach.

Again, caution must be exercised with respect to the permanency of the recommended control approach. In particular it must be kept in mind that all factors are not known relative to atmospheric interactions. Because of this lack of information it is most difficult to develop a comprehensive systems approach to this problem. A good example of the need for caution was discovered when the changes in the Los Angeles atmosphere between 1962 and 1967 were investigated.14 During this 5-year period, in the May through October months, the 6 to 9 a.m. hydrocarbons decreased by 4 percent and the nitrogen oxides increased by 25 percent. Additionally the maximum daily 1-hour average oxidant decreased by 11 percent. Focusing attention on only the decrease in maximum oxidant level, however, does not describe all the changes. Thus this maximum oxidant decrease was associated with an increase in the number of days on which the maximum oxidant exceeded 0.1 ppm. The result is that a trade-off has been made, i.e., an 11 percent decrease in the maximum for a 10 percent increase in dosage. This latter increase in dosage, and to a large extent the decrease in maximum, may be related to the altered ratio of HC to NO_x. Thus control approaches which suggest allowing increases in ambient levels of nitrogen oxides are definitely subject to question. As a result of this and other examples it is apparent that we must be always alert to the atmospheric stiuation on a year-to-year basis. Furthermore, we must be prepared to modify any adopted control approach as new information becomes available.

In addition to these examples of lack of information, there are other contributing factors which may demand changing any projected emission values at some time in the future. During the next five years, for example, we will re-examine the Photochemical Oxidant and Hydrocarbon Criteria documents. In the intervening years it is quite possible that more precise measuring methods may be developed. Thus the oxidant criteria may become ozone criteria and the present non-methane hydrocarbon measurement may be stated in terms of reactive hydrocarbons. Such changes will, of course, affect the emission goals as calculated in this presentation.

Summary of Calculations Relative to Motor Vehicle Emission Goals Designed to Produce a Health-Related Acceptable Air Quality

Calculation of the motor vehicle emission goals for CO, HC, and NO_x requires application of two equations. The first of these equations permits a calculation of the fractional reduction in ambient concentrations necessary to achieve a specific health-related air quality.[15] The equation is stated in terms of present air quality, desired air quality, background concentrations, and the projected growth in emissions. Mathematically stated, it has the form:

$$R = \frac{(GF)}{(GF)} \frac{(PAQ) - (DAQ)}{(PAQ) - (B)} \qquad (1)$$

where R is the calculated fractional reduction required; GF is the emission growth factor; PAq is the present maximum air quality; DAQ is the desired air quality; and B is the background concentration. The second equation employs the result from Equation 1 to restate the air quality reduction in terms of mobile emission rates. Mathematically expressed, this second equation has the form:

$$(PER) \quad (1-R) = DER \qquad (2)$$

where PER is the percent emission rate giving rise to the present air quality (PAQ); the expression 1-R is the statement of how much of the emission can be allowed without exceeding the desired air quality. The multiplication, i.e., (PER) (1-R) thus yields the desired emission rate (DER) consistent with the desired air quality (DAQ).

In order that the calculated emission goals be protective of health, certain principles must be adopted relative to the values inserted in Equations 1 and 2. Thus, present air quality in Equation 1 must be representative of the worst situation observed in the entire United States. For HC and NO_x values to be associated with health effects of oxidants and eye irritants, the worst situation presently occurs in Los Angeles.[16] (The 6 to 9 a.m. maximum average concentrations observed during the high oxidant potential season were 5.3 ppm for non-methane HC and 0.62 ppm for NO_x.) Los Angeles also had the highest yearly 1-hour average NO_2 concentrations (0.69 ppm); thus, when the health effects of NO_2 are being considered, air quality levels in Los Angeles are used. With respect to CO health effects, however, the city of Chicago, with a maximum 8-hour average of 44 ppm, represents the worst known case in the United States.[16] In terms of growth factors, we must again assume the maximum predicted growth in order to provide adequate health protection. For a time period from 1967 to ten years beyond the year of application of these motor vehicle emission goals, the maximum predicted growth factor for mobile emissions must be used. Again, to afford the maximum health protection, the background concentrations used in Equation 1 must be the maximum supported by scientific inquiry. For these purposes a value of 1 ppm was used for CO,[10] 0.1 ppm for HC,[11] and 0.004 ppm for NO_x. The lowest supportable values for the desired air quality, as derived from the appropriate criteria documents, were used in Equation 1 and have been presented in Table I.

For the present emission rate (PER), we have related derived emission rates to a 1967 rate of 82.6 grams per miles (g/mi) for CO, 14.84 g/mi for HC, and 5.93 g/mi for NO_x.[17] The HC and NO_x rates are representative of the 1967 vehicle mix in Los Angeles, while the CO rate is representative of the 1967 vehicle mix in Chicago.

The one factor which this treatment does not speak to is the possible effect of the failure of vehicles to meet the stated emission goals when they become standards. Thus, surveillance data from vehicles presently in consumer use show current Federal standards are exceeded on the average by 13 percent in terms of CO emissions and 25 percent in terms of HC emissions.[18] This, however, is a difficult factor to take into account because it may change in the future. If we ignore such past failures, however, we are in essence announcing that before these emission goals are established as standards we will have devised methods that will force automobile manufacturers to provide control devices which will enable vehicles in consumer use to meet required emission standards. The projected emission limitations derived here will assume that devices and control systems in light duty vehicles in use continue to exceed the standards by the stated percentages.

To proceed to the calculation of specific mobile emission goals needed to achieve desired air quality goals, it now becomes necessary to designate a target year for implementation of the required mobile emission restrictions. Selection of a target year should be based upon the evaluation of many important factors such as: the need to meet the desired air quality goals at the earliest possible time; the state-of-the-art of control technology; and future advances in our understanding of atmospheric photochemical processes. It is beyond the scope of the paper and the competencies of the authors to evaluate all of the above factors and their interactions in detail. Thus, to avoid becoming embroiled in high-level policy questions which still remain to be resolved, we have arbitrarily selected 1980 as the target year on which to base our calculations. In the event the decision is made to select an earlier year for implementation it will be an easy matter to adjust the calculations accordingly. Of course, the earlier the calculated mobile emission restrictions can be put into effect, the earlier we will be able to achieve desired air quality goals and the less stringent will be the necessary mobile emission limits. In accordance with the above decision we used a 1967-1990 mobile emission growth factor.[17,19]

In reviewing the required information items it will be noted that the percentage contribution of current mobile sources to existing air quality, the projected mobile growth rate, and the background concentrations of each pollutant are the least well-defined. Thus, background concentrations of HC are not delineated with any degree of accuracy. The same can be said about growth rates of mobile emissions. Furthermore, the percentage contribution of mobile emissions to total emissions must at this time be based on logic rather than known fact. In the case of Los Angeles there seems to be general agreement that in the downtown area the smog-season emissions of CO, HC, and NO_x are mainly from mobile sources. Consequently, the calculated emission goals were based on the latter assumption. A similar assumption was made relative to the Chicago data, i.e., that the bulk of the ambient CO concentrations steam from the mobile source. If this assumption is incorrect, it follows that those stationary sources which contribute significantly to the downtown area concentrations of CO, HC, and NO_x must be controlled to the same degree as motor vehicles. To the extent that these sources are uncontrollable the restriction of the motor vehicle would have to be increased.

Table II contains a summary of the numerical values used in calculating 1980 mobile emission goals. Figures 4, 5, and 6 express the results of the calculations for desired mobile emission rates for CO, HC, and NO_x, respectively, as a function of the desired air quality for each pollutant without taking into account the potential deterioration of the control measures for CO and HC. In these figures a range of values above and below the Table 1 values of desired air quality are shown since the exact health-related values are still under study and may be changed in the future. Table III contains the 1980 motor vehicle emission goals for CO, NO_x, and HC required to achieve the desired air quality after taking presently observed CO and HC control measure deterioration into account.

Table II. Numerical Values of Parameters used in Calculating Mobile Emission Goals[9-11,16,17,19]

Parameter	CO	HC	NO_x	NO_2
1967 maximum air quality values related to direct health effects	51 mg/m^3 for 8 hr, Chicago	-	-	1300 $\mu g/m^3$ for 1 hr, (0.69 ppm), Los Angeles
1967 maximum air quality of oxidant precursors, (6 to 9 a.m. average)	-	3.5 mg/m^3 as CH_4 (5.3 ppm C), Los Angeles	1170 $\mu g/m^3$ as NO_2 (0.62 ppm), Los Angeles	-
1967 average emission rates for all motor vehicles	82.6 g/mi	14.84 g/mi	5.93 g/mi	-
Maximum background concentration	1 mg/m^3 (1 ppm)	0.1 mg/m^3 as CH_4 (0.1 ppm)	8 $\mu g/m^3$ as NO_2 (0.004 ppm)	8 $\mu g/m^3$ as NO_2 (0.004 ppm)
Maximum growth factor of mobile emissions	2.18	2.18	2.18	2.18
Air Quality Goals	10 mg/m^3 for 8 hr (9 ppm)	125 $\mu g/m^3$ of oxidant (0.06 ppm), 1-hour average	125 $\mu g/m^3$ of oxidant (0.06 ppm), 1-hour average	190 $\mu g/m^3$ for 1 hr (0.1 ppm)

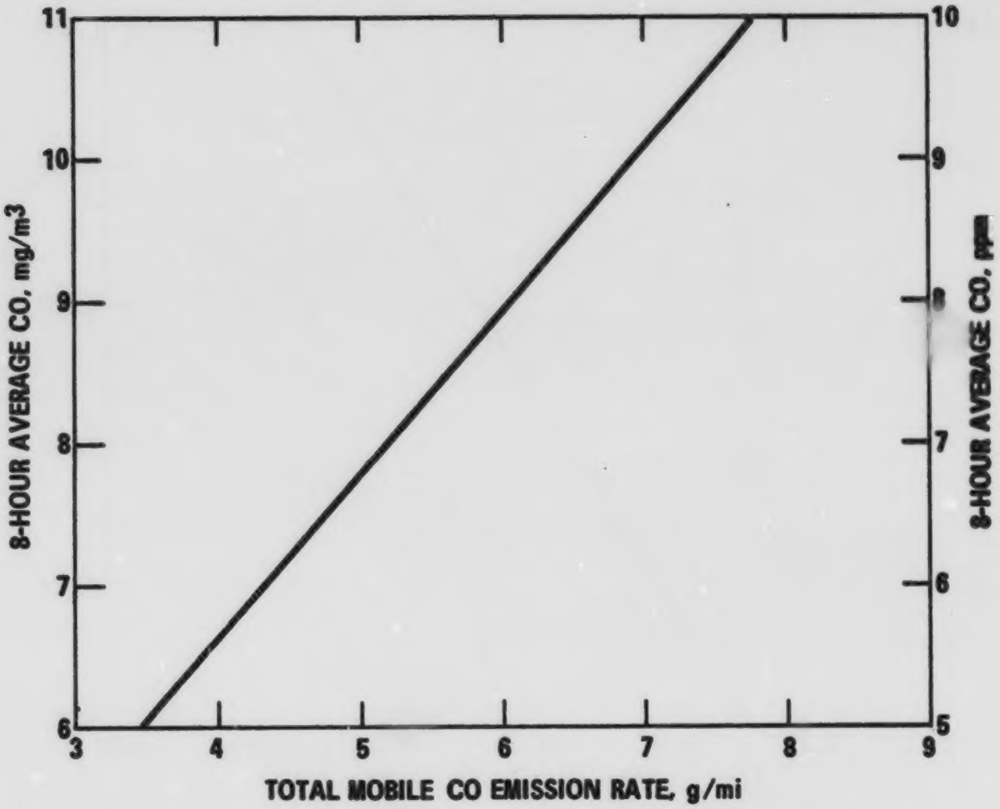

Figure 4. Relationship of 8-hour average CO values to mobile emission rates.

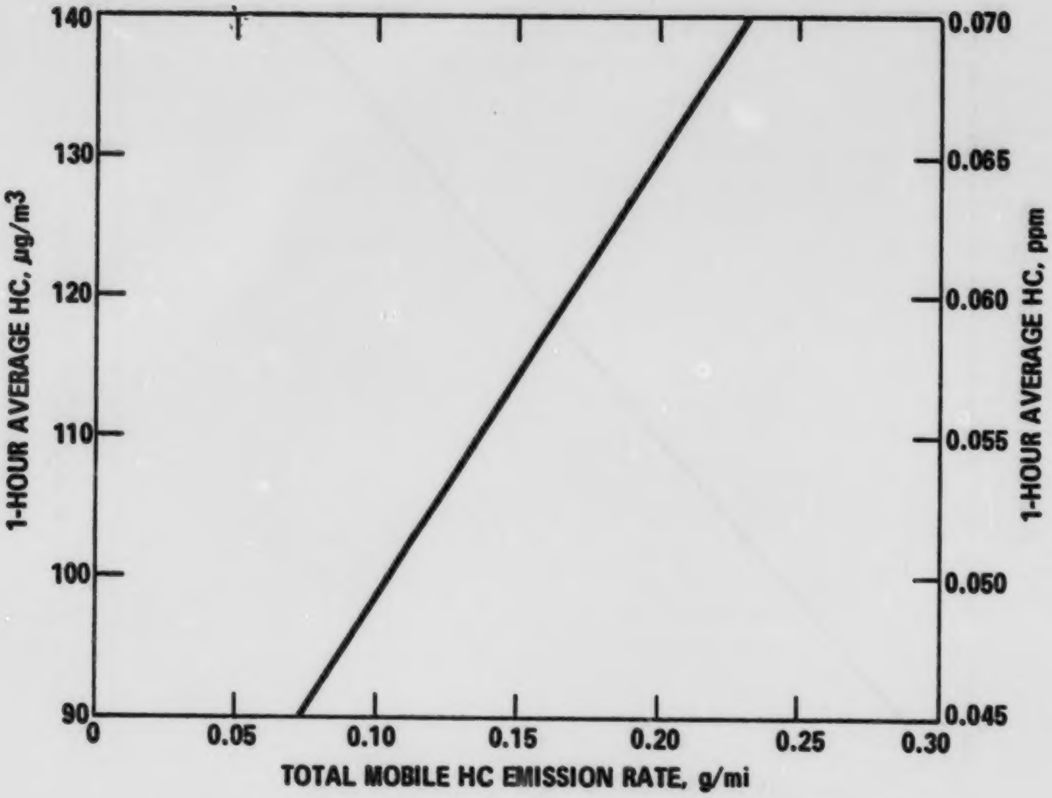

Figure 5. Relationship of HC mobile emission rate to 1-hour average oxidant.

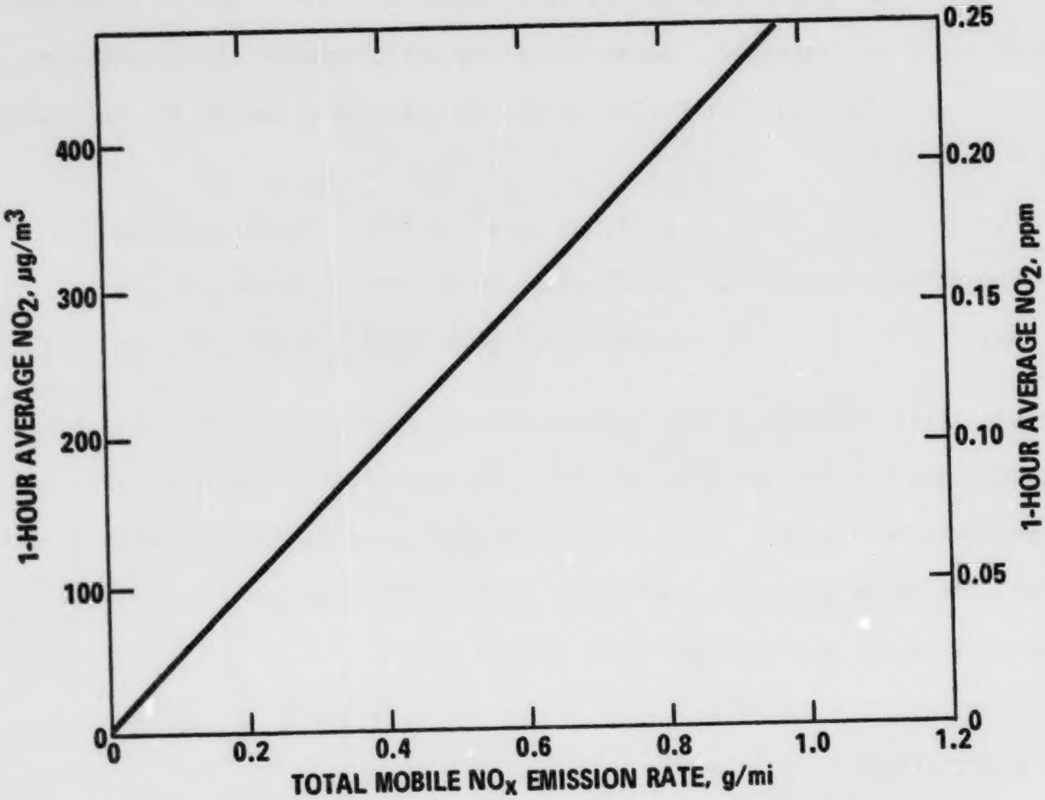

Figure 6. Relationship of 1-hour ambient NO$_2$ levels to mobile NO$_x$ emission rates.

III. Summary

The historical background of the development of Federal mobile emissions standards has been presented. Based on issued and planned Criteria documents, desired air quality goals for health protection have been set at the following levels:

CO	$\leq$ 10,000 μg/m^3 (9 ppm) - 8-hour average
Photochemical oxidants	$\leq$ 125 μg/m^3 (0.06 ppm) - 1-hour average
NO_2	$\leq$ 190 μg/m^3 (0.10 ppm) - 1-hour average

The complexities inherent in the control of photochemical oxidants have been discussed and a control strategy involving joint control of hydrocarbons and nitrogen dioxides adopted. Using a modified roll-back calculation with a 1967 baseline, the following 1980 total motor vehicle emission goals by which to achieve desired air quality goals have been derived:

CO	6.16 g/mi
Hydrocarbons	0.14 g/mi
NO_x	0.40 g/mi

The calculation of future motor vehicle emission goals should be a continuous process with new data being used as they become available.

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Chapter III

THE ADVANCED AUTOMOTIVE POWER SYSTEMS PROGRAM

by

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Introduction

The Advanced Automotive Power Systems Program is an outgrowth of concern by our nation, led by President Nixon, that improvement of air quality in the United States is markedly dependent on the elimination of the automobile as a significant source of air pollution. In the United States, the internal combustion engine in our automobiles contributes about half of the total pollutants from all sources to our air environment. Figure 1 illustrates the automobile's share of the blame in the United States. Based on 1970 estimates, nearly 70% of the carbon monoxide from all man-made sources comes from the automobile, also 40% of the hydrocarbons and about 40% of the nitrogen oxides appear from this source. Less than 1% of the particulates and less than 0.10% of the sulfur oxide emissions nationwide are due to this source.

One result of the nationwide concern over the automobile has been the tightening of Federal exhaust emission standards that must be met on new automobiles. This tightening of the standards is apparent by inspection of Figure 2. This Figure shows the Federal exhaust emission standards through 1973 for the three major pollutants and estimated values for 1975/76. The final values for these latter years have not been finalized, as yet. By model year 1975, a 90% reduction in hydrocarbons and carbon monoxide from the 1971 levels are shown, and, by 1976, the nitrogen oxides levels will be equivalent to a 90% reduction from the uncontrolled levels measured for the 1971 model year vehicles. There is considerable doubt that the conventional spark-ignition gasoline-fueled reciprocating internal combustion engine can be 'cleaned up' adequately to meet all of these forthcoming standards. As a result, alternate power systems which are inherently clean when compared with the conventional engine must be considered as potential replacements.

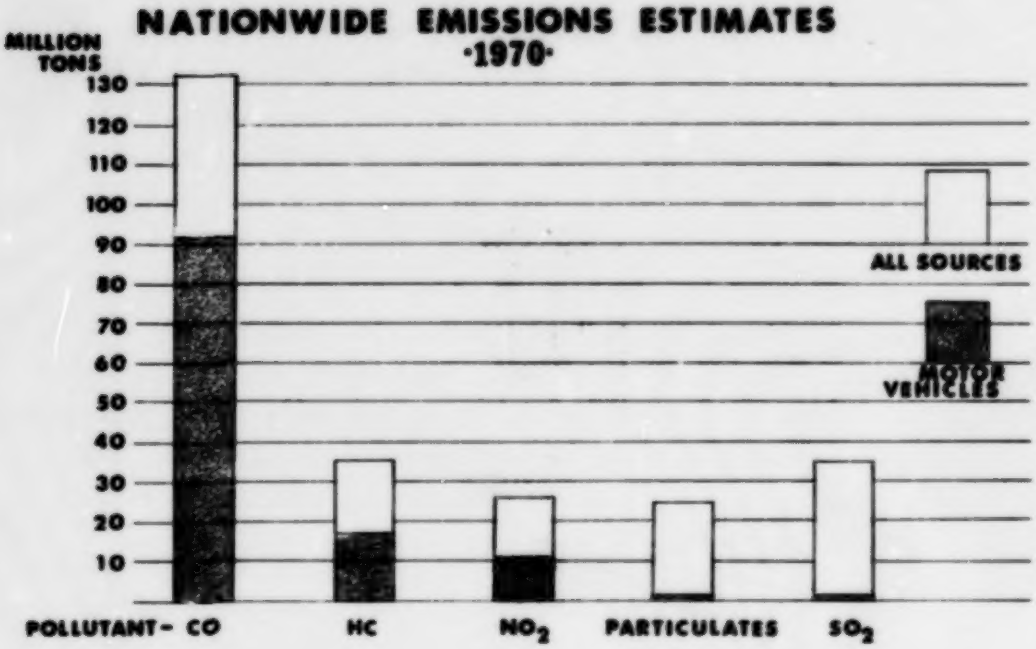


Figure 1.

FEDERAL MOTOR VEHICLE EMISSION STANDARDS

grams/mile

	1971	1972	1973	1975	1976
HC	4.6	3.4	3.4	0.46	0.46
CO	47.0	39.0	39.0	4.7	4.7
NO_x	–	–	3.0	3.0	0.4 – 0.6

Figure 2.

Bases for the Program

A panel of scientists originating in the Executive Office of the President was formed in 1969 to evaluate industry efforts in our country to develop alternate power systems. The conclusion reached by the Panel was that industry was not making a serious effort to make such developments. Lacking sufficient industry interest, a Federal research and development program was recommended. This program was announced by President Nixon on February 10, 1970 (see Figure 3) and the program was underway by July. In addition, in his February message the President announced another Federal program to stimulate industry by providing financial incentive to groups who independently develop their own alternate power systems. This latter program is called the Clean Car Incentive Program. Thus, the two parallel approaches to develop alternate power systems appear as in Figure 4. The total package consisting of the Research and Development Program and the Incentive Program is called the Advanced Automotive Power Systems (AAPS) Program.

What are the goals of the Advanced Automotive Power Systems Program? The program is intended to provide insurance to our nation that if a practical and virtually pollution-free power system could be developed, this development would culminate in a demonstrated system by 1975. Achievement of this goal can be accomplished by demonstration of a complete power system either developed directly under Federal government monies or, by similar type demonstrations whose development is sponsored by industry, perhaps as a result of stimulation brought about by the existence of the Research and Development Program or of the Incentive Program, or both.

In addition, the AAPS Program will provide important information to the Administrator of the Environmental Protection Agency on

EXCERPT FROM MESSAGE ON ENVIRONMENT —
FEBRUARY 10, 1970

"... with the goal of producing an unconventionally powered, virtually pollution free automobile within five years."

— PRESIDENT NIXON

Figure 3.

DEVELOPMENT OF ALTERNATE POWER SYSTEMS APPROACH

TWO PARALLEL PATHS

FEDERALLY SPONSORED R/D

INCENTIVE PROGRAM FOR PRIVATE DEVELOPMENT

Figure 4.

the technical feasibility of meeting the stringent 1975/76 emission standards with use of alternate power systems. Such information aids in judgments made by the Administrator on possible waiver requests to postpone enforcement of the standards.

Research and Development Program

First, the research and development part of the Program will be discussed. What are candidates for development in this Program? There are five types of power systems that were initially part of the Program when it began in July 1970. These include the gas turbine, two kinds of hybrids--heat engine/electrics and heat engine/flywheel systems--Rankine cycle powerplants and the all-electric system. There are two additional systems, the stratified charge engine and the Diesel engine which are serious contenders for candidacy. The particular versions of these last two systems under consideration are reasonably well-developed and are being considered as candidates because of their nearer term potential for meeting the 1975/76 emission standards when mass produced. Thus, it is these two engines which we expect will provide important information on whether alternate systems can meet the standards and this information will be available very early in the Program.

Figure 5 lists all of the candidate systems and the stratified charge and Diesel. In general, the candidates were selected because the technology to improve, to improvise, and to further develop each, exists in abundance in the United States. In addition, independent research sponsored by private industry in the United States had been underway for some time on most systems selected. Thus, the interest to further develop the systems was there. Additional candidate systems may be brought into the Program. We are continually reassessing new developments on other systems including developments made within our country and worldwide.

It is pointed out that the Stirling cycle powerplant has been seriously considered for candidacy since the Program work began. The considerable strides made in the Netherlands and Sweden in making the Stirling cycle powerplant a competitive system have been followed closely by our staff and our Technical Advisory Committee. However, we are not ready as yet to bring this system into the Research and Development Program.

The all-electric is shown apart from the other candidates in Figure 5. We do not see the complete development of the all-electric system early enough to meet the 1975 date because the type of batteries with adequate power and energy densities needed to compete in performance and cost with the conventional engine are presently in the basic research stage. The battery systems of special interest herein are the high temperature alkali-metal battery systems such as Lithium-Sulfur and Sodium-Sulfur. We recognize that there are several types of batteries in use today for all-electric automobiles and small vans. However, in order to meet the driving requirements of the American public, the development of completely new and more powerful batteries is needed. As a result of these considerations the all-electric car is not a candidate to meet the 1975 goal, rather we see its development by 1978. So, one may ask why is this system in the Program? It is in the Program because eventually these new and better battery systems must be developed. The cost of development is very high as is the risk. For industry at the present time, the costs and risks appear to outweigh the ultimate advantage of having these batteries available. By supporting the research and early development with government monies we see a point being reached where industry will pick up the further development of these batteries.

On what basis will any of these propulsion systems be brought through the development phases into complete system hardware?

CANDIDATE POWER SYSTEMS

GAS TURBINE

HEAT ENGINE/ELECTRIC HYBRIDS

2nd GENERATION ICE's — STRATIFIED CHARGE ENGINE
DIESEL ENGINE

RANKINE CYCLE

HEAT ENGINE/FLYWHEEL HYBRIDS

ALL ELECTRIC

Figure 5.

The criteria which we consider are summarized in Figure 6. Performance requirements are defined for the system and for individual components. Based on the designs formulated and the tests conducted, we must have confidence that these requirements can be met. The characteristics of the largest selling model type automobile in the United States for year 1970 form the basis for engine weight and volume constraints, road performance requirements and fuel economies. Of course the exhaust emissions must meet or do better than the emissions standards of 1975/76. The safety characteristics of the automobile with the new engine are another important consideration. For example, it is desirable that the working fluid in the Rankine cycle engine be non-toxic and non-flammable. The flywheel design must be such that any structural failure of the flywheel itself not lead to catastrophic failure outside of the confines of its casing. The socio-economic impact noted on the slide relates to the criteria that engine exhaust odor and noise must not be ojectionable according to contemporary standards and that the system cost to build in-quantity production be competitive with the conventional engine with its exhaust treatment devices installed on the latter. National impact is listed last, yet, it ranks as number one on anyone's list of criteria. As one example of how we evaluate the impact, let us consider impact of the all-electric automobile.

Figure 7 shows the projected power generating capacity in the United States beyond year 2000. In some of the most densely populated areas of our country, such as New York City, we find that the demands for electricity, at times, are not being met by the supply. This is a city without any appreciable electric car popultion. On a nationwide basis, if all the vehicles on the road today were all-electrics, the total demand for electricity would exceed the existing capacity to supply by

POWER SYSTEM ACCEPTANCE CRITERIA

- SPECIFIED SYSTEM AND COMPONENT PERFORMANCE REQUIREMENTS
- VERY LOW EXHAUST EMISSIONS
- SAFETY
- SOCIO-ECONOMIC REQUIREMENTS
- NATIONAL IMPACT

Figure 6.

PROJECTED POWER GENERATING CAPACITY IN THE UNITED STATES

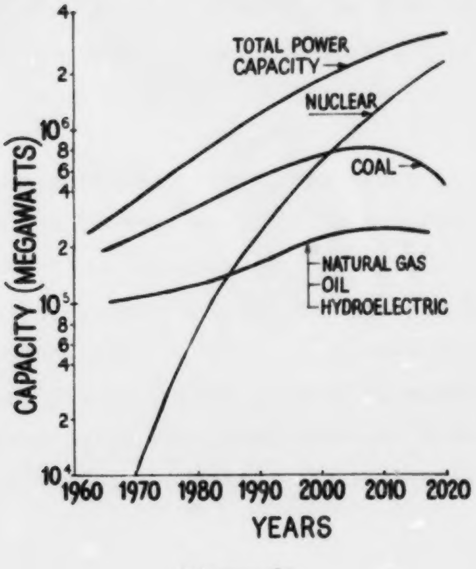


Figure 7.

about 40%. In addition, the technology to control sulfur oxides from fossil fuel burning power stations is still being developed. Further, the nuclear source curve shown in Figure 7 is very optimistic and since drawing this Figure, one year ago, our experience has been that as time goes on this cross-over point will move farther to the right. Therefore, if rechargeable alkali-metal batteries were fully developed right now and all-electric automobiles produced in-quantity we would aggravate our existing electrical power capacity problem and, in addition, temporarily transfer the major sources of pollution from a mobile to a stationary source. So then these are some of the factors that we consider in judging whether a candidate powerplant should go beyond the paper study phase and into hardware, and, of equal importance, these factors influence the timing of certain research work and the pace of development of some candidates.

Individual Candidates and Their Status

Now let's take a closer look at the candidates and their status. Each candidate was at a different state of development when the Program began, and depending on our success in achieving technological breakthroughs, each may enter the system hardware phase at different times during the next five years. The first 18-month phase of the Program is intended to be a period of evaluation wherein the systems and their components are to be designed, the critical components bench tested and decisions made on whether to proceed to first generation hardware.

Rankine Cycle Engine. Consider the Rankine cycle engine as illustrated in Figure 8. In this engine, there is an external combustor and an enclosed working fluid which is heated, expanded to do work, then condensed to a liquid, with the fluid being continuously recycled. There are three types of Rankine

BASIC RANKINE CYCLE ENGINE

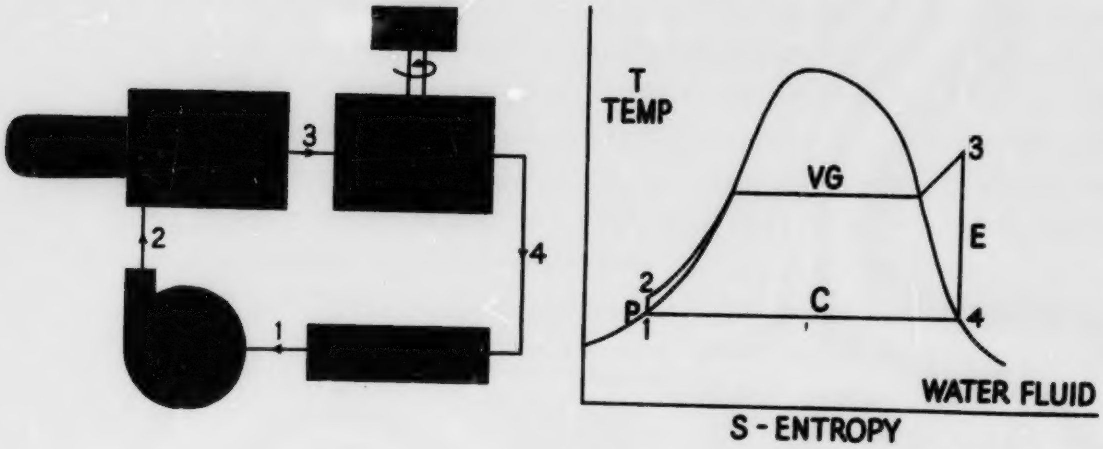


Figure 8.

CANDIDATE RANKINE CYCLE POWER PLANTS

- ORGANIC WORKING FLUID - RECIPROCATING EXPANDER
- WATER BASE WORKING FLUID - RECIPROCATING OR A ROTARY EXPANDER
- ORGANIC WORKING FLUID - TURBINE EXPANDER

Figure 9.

systems which are presently in the design and component test phase as shown in Figure 9. The work on the organic working fluid reciprocating expander system is summarized in the technical paper presented in this meeting by a representative from Thermo-Electron Corporation where this work is being conducted in our country. Work on the other versions of Rankine cycle systems noted on the Figure are just beginning. As we see them, the problem areas associated with a practical design of the Rankine cycle appear on the next slide. The problems appear mainly in the inefficiency of components and complexity of the control system and, of course, with its exhaust emissions. A representative of Solar Division of International Harvester Company in San Diego, California, one of three firms conducting government sponsored research and development on improved combustors will summarize the work of one of the high efficiency combustor systems at this meeting. The next Figure summarizes a schedule of the planned work on the Rankine cycle. Complete system development for the three types mentioned earlier with parallel research on the components is shown. We point out that the three systems contractors are designing complete systems including all components. An integral part of this effort is to define the design requirements that each component must meet. For example, the temperature of the condensate, maximum and minimum flow rates and heat release rates are part of the condenser design requirements. The systems contractor and the backup component research contractor are designing a condenser to meet the identical requirements. The condenser research contractor will design and verify in test his condenser designs for each of three systems. The first generation system hardware for any of the three systems may have components made by one or more of the component research contractors, depending on whose component design is the most efficient with cost to produce being one of several important

RANKINE CYCLE ENGINE PROBLEMS

- •CONDENSER SIZE WT.
- •BOILER SIZE WT.
- •CONTROL COMPLEXITY
- •ENGINE EFFICIENCY
- •MINIMIZE EMISSIONS
- •FREEZING (water)
- •LUBRICATION
- •FEEDPUMP
- •SEALS (non-water)
- •VALVING DESIGN

Figure 10.

RANKINE CYCLE ENGINE PROGRAM

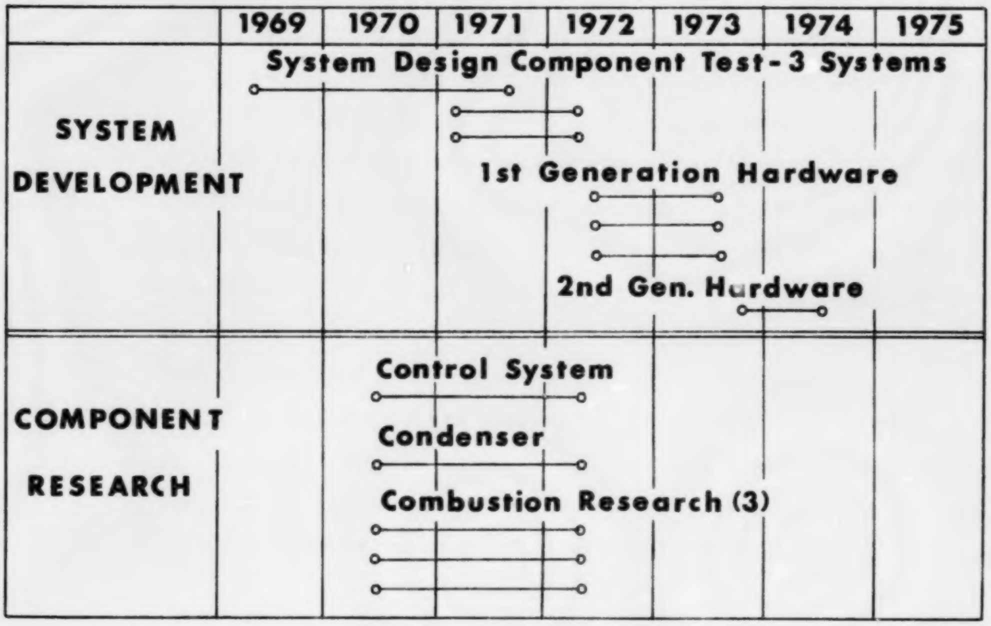


Figure 11.

elements applied in that judgment. Lastly, you can see from Figure 11 that delivery of the first prototype engines could be as early as 1973.

Gas Turbine. More work has been conducted by industry in our country on the gas turbine than on any other candidate with an unconventional engine for the automobile application. For the gas turbine we are focusing our efforts on solving some of the problems which have plagued past attempts to get the gas turbine on the road. Some of these problems are noted in Figure 12. These problems include the need for: developing manufacturing techniques for mass producing turbine inexpensively, for improving the part load, fuel economy, combustion research to improve combustor design and increasing system reliability. Actually, if we operated our automobile in the same way that commercial jet aircraft are operated, namely, running most of the time at constant speed, many of these problems would disappear. This gas turbine program is oriented toward solving problems, and if successful, is intended to stimulate industry to apply the results of this research to their own turbine designs. As we see it, the demonstration of the turbine in the automobile probably will be performed by industry. It is because industry is so close to practical hardware on this system that our approach here emphasizes problem solving rather than a government sponsored demonstration.

One of the big unknowns in getting the turbine on the road is knowledge of the cost to mass produce this system; therefore, early recognition of what are the high cost components of the system is needed so that fruitful manufacturing research can be properly directed. A representative of Pratt & Whitney Aircraft from the United States will present findings on this subject in this meeting. That firm is on the team of contractors in this program.

Hybrids. Now, for the hybrids, the heat engine/electric and the heat engine/flywheel. The heat engine/electric hybrid consists of a small size engine such as in a Volkswagen, and an array of inexpensive lead-acid batteries. The manner in which these two power sources are arranged is illustrated in Figure 13. The same series versus parallel configurations can be applied where the flywheel replaces the battery system. The system is designed to extract power from the heat engine alone, or from both sources, heat engine and battery, at the same time. We find that the parallel hookup is best where the conventional internal combustion engine is used as the heat engine and a series configuration appears best where a small gas turbine is used. In either configuration the system operates by running the heat engine at a constant speed--say equivalent to 40 mph road speed, and, where vehicle accelerations are required, the additional power comes from the battery system. Above a 40 mph road speed the engine speed slowly converges on the new steady-state level demanded by the driver with additional power required during slack time provided directly from the battery system. There are several potential advantages which this hybrid concept offers. One is that the engine speed range is relatively small with the attendant ease of control of exhaust emissions under such a condition. Another advantage is that the very high road performance for a standard size American automobile, of approximately 4,000 lb. weight, can be demonstrated using a relatively small and inexpensive heat engine. A basic problem with this type system appears in its relative complexity, higher cost and the larger system volume required with use of two power sources. Development of improved lead-acid batteries to accommodate the rapid charge-discharge characteristics needed for this mode of operation is now underway in the Research and Development Program.

In our analysis of this type hybrid many types of heat engines have been considered by our contractors. Two types were noted

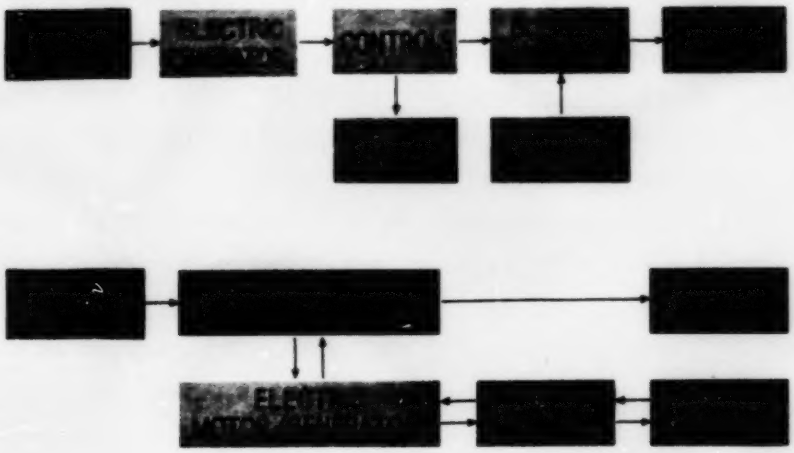
GAS TURBINE ENGINES

MAJOR PROBLEM AREAS

- MANUFACTURING COST
- FUEL ECONOMY
- EMISSIONS - NO_x REDUCTION FOR 1980 GOALS
- RELIABILITY

Figure 12.

HYBRID SCHEMATIC



SERIES HYBRID DRIVES WITH BATTERY ENERGY STORAGE

PARALLEL HYBRID DRIVES WITH BATTERY ENERGY STORAGE

Figure 13.

earlier. Two firms in our country are conducting the bulk of our work on this hybrid. One firm is the Aerospace Corporation. A member of the Aerospace Corporation team will report on their work at this meeting.

As stated earlier, the heat engine/flywheel system would function in a similar manner to the heat engine/electric with the battery replaced by a mechanical storage device, namely, the spinning flywheel. We have progressed on this system from parametric analysis of many practical flywheel materials such as shown in Figure 14 and numerous practical configurations as shown in Figure 15 to the design and fabrication of specific flywheels for cars.

Today we are at the point where two flywheels are now being tested to verify predicted energy and power densities. One of the flywheels is fiberglas and the other fabricated of 4340 steel. Work on the configuration of the key component of this system, the transmission, will begin shortly. To give you a feel for some characteristics of the flywheel: for a full size family car, the flywheel is made of 4340 steel in a constant stress configuration weighing 42 lbs. and operating at a maximum speed of 24,000 revolutions per minute.

All-Electric Car. Alkali-metal battery research and development for the automobile application has been underway for more than a year at Argonne National Laboratories. Their work was reported on in this meeting. We anticipate that the 'proof-of-principle' for the high temperature lithium-sulfur system will be demonstrated on single and multiple cells within the next 18 months. This 'proof-of-principle' refers to the experimental demonstration that the energy density and power density desired for a full-sized battery can be obtained on an elemental cell basis. Once this proof-of-principle has been achieved, the battery work will move into a development phase, first with a goal of a 2 kw battery, then a 5 kw battery and then a 20 kw battery system.

FLYWHEEL MATERIALS

MATERIAL	DENSITY (ρ) LBS/IN^3	POISSON'S RATIO (υ)	ULT. TENSILE (F_{tu}) KSI	YIELD TENSILE (F_{ty}) KSI	REC. WORKING STRESS (σ) KSI	$\frac{\sigma}{\rho}$ ($\times 10^6$)	MATL COST ($/LB)	NORM COST
18NI-400 (MARAGING STEEL)	0.289	0.26	409	400	260	0.900	2.25	4.20
18NI-300 (MARAGING STEEL)	0.289	0.30	307	300	200	0.692	2.25	5.47
4340 STEEL	0.283	0.32	260	217	130	0.459	0.60	2.20
1040 STEEL	0.283	0.30	87	58	36	0.127	0.30	3.97
1020 STEEL	0.283	0.30	68	43	25	0.088	0.30	5.73
CAST IRON	0.280	0.30	55	37	20	0.071	0.30	7.10
2021-T81 (ALUMINUM)	0.103	0.33	62	52	26	0.252	0.53	3.54
2024-T851 (ALUMINUM)	0.100	0.33	66	58	35	0.350	0.50	2.40
6AL-4V (TITANIUM)	0.160	0.32	150	140	82	0.512	4.00	13.13
"E" GLASS	0.092	0.20	130	-	65	0.706	0.42	1.00
S-1014 GLASS	0.067	0.20	250	-	128	1.910	2.40	2.11

Figure 14.

FLYWHEEL GEOMETRIES

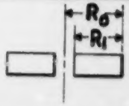
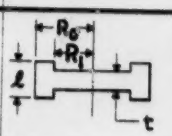
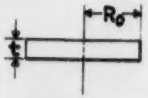
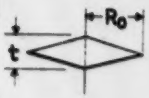
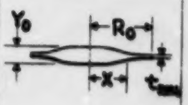
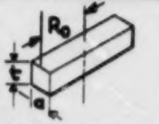
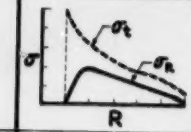
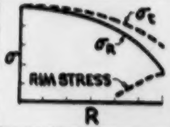
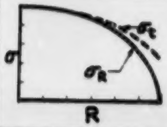
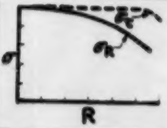
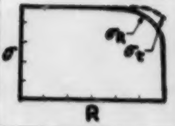
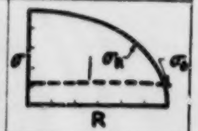
	PIERCED DISC	RIM	SOLID DISC	CONICAL	CONSTANT STRESS	LOG OR BAR
SHAPE						
STRESS MAP						
VOLUMETRIC EFFICIENCY (FW& MIN. WT. HOUSING)	30.4%	20%	30.4%	62%	80%	6%

Figure 15.

Stratified Charge Engine. As mentioned earlier, the two candidates which we are seriously considering for entry into the Program are the stratified charge engine and the Diesel. The stratified charge engine is a spark-ignited gasoline fueled internal combustion engine with many hardware characteristics of the conventional engine. Differences appear mainly in the combustion chamber design, use of fuel injection, and in the resulting combustion process. In one version of this engine the fuel is injected into the cylinder as the spark appears and therefore the fuel burns as it enters. The fuel burns initially in an over-rich condition with fuel injectors designed to permit fluid swirl at the top of the cylinder. The burning expands into the lower portions of the cylinder and, overall, the resultant combustion products are similar to those obtained using a very lean mixture. Lean burning is very desirable from the emissions viewpoint. The initial work on this engine was sponsored by the U.S. Army Tank-Automotive Command in the state of Michigan. The measured exhaust emission levels for the stratified charge engine installed in jeeps and employing a catalytic muffler are below the standards for hydrocarbons and carbon monoxide set for 1975 but further work must be conducted to reduce the nitrogen oxide emissions. Several generations of development have been funded and if this system is brought into the Program, the work will emphasize achievement of reductions in nitrogen oxide emissions and then fleet testing of this well-developed system.

Diesel Engine. The Diesel engine is not commonly used in American made automobiles, mainly because it is heavy and costs more to manufacture compared with the conventional Otto cycle engine. If the Diesel is brought into the Program, emphasis will be placed on furthering the development of a low compression ratio Diesel with a high swirl and prechamber design. Exhaust emission levels for hydrocarbons and carbon monoxide which are lower than the 1975 standards have been shown for this type engine without resorting to a catalyst. The measured nitrogen oxide levels on the first generation engine are relatively low but are within reaching distance of the 1976 standard. Work on this type Diesel will concentrate on nitrogen oxides reduction, and on performance durability and driveability testing.

Federal Clean Car Incentive Program

The goals of the Research and Development Program, the near candidates and our approach to further development of each have been discussed. As stated earlier, there is another important program underway in our nation to develop a virtually pollution free automobile using an alternate power source. This work consists of efforts on the part of industry itself with financial incentive provided by our government to inspire independent work. This describes the Federal Clean Car Incentive Program. The intent of the Program is to stimulate the private sector to produce a virtually pollution-free automobile. The program provides a market for large and small auto and non-auto manufacturers who often possess new and unique approaches toward engine designs. However, in the past, they have lacked incentive to further independent development.

The Program description is summarized in Figure 16. After successfully passing stringent emissions and performance testing, first on a leased Prototype car, then on 10 purchased copies of the prototype for Demonstration, the successful engine system will be further tested after procurement of 300 vehicles. If the low emissions levels are maintained and road performance satisfactory, the car then receives certification as a low-pollution vehicle, and this certification is significant. The certified vehicles which make their way through the Program will be favored by purchase in quantity for Government fleet use. Premiums as high as 200% over the basic price normally charged to the government are permitted for the successful developer.

ELEMENTS OF THE FEDERAL INCENTIVE PLAN

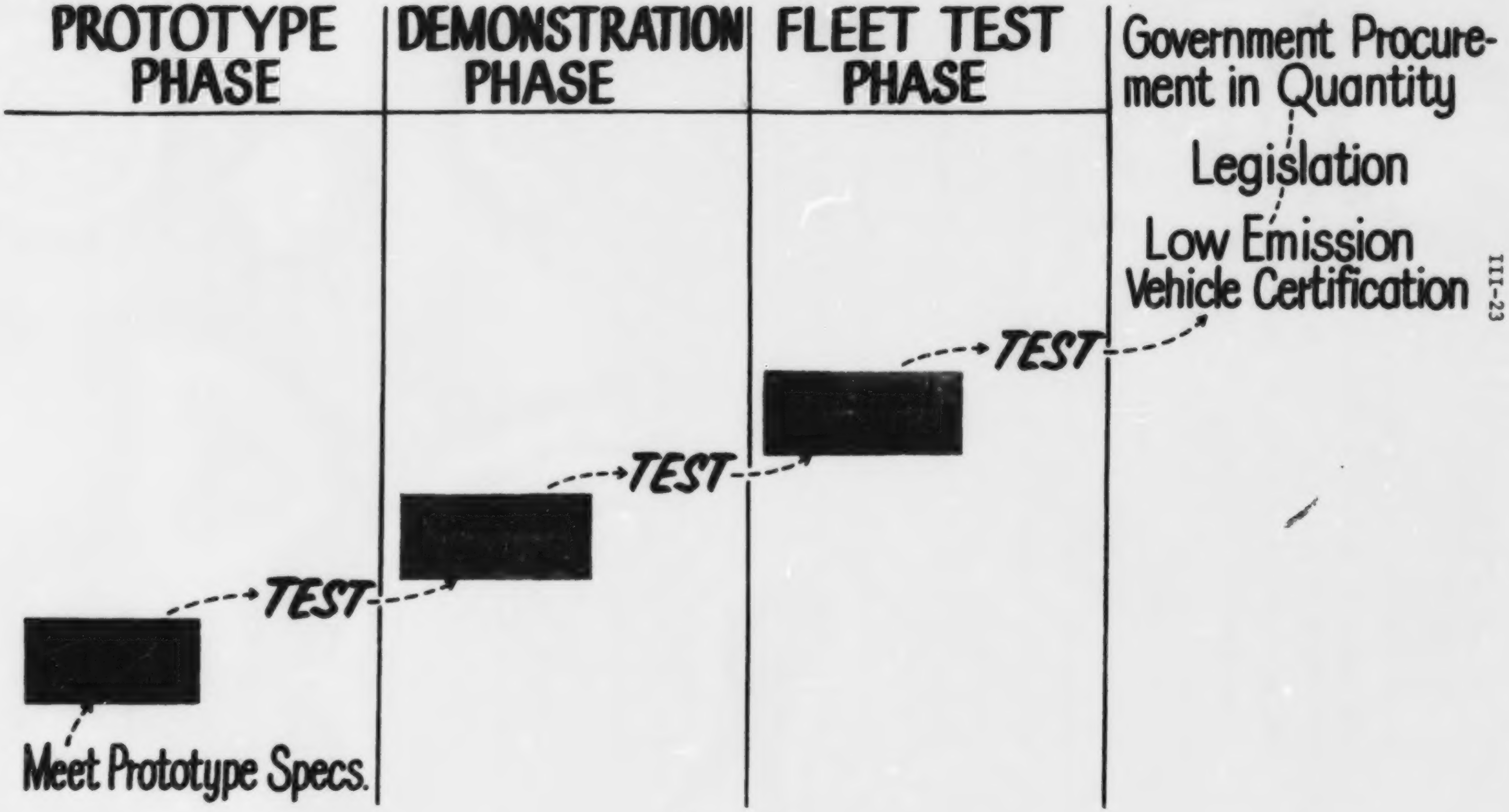


Figure 16.

The legislation to accommodate such purchases appears in the Amendments to the Clean Air Act, recently passed by our Congress and signed by our President. We point out that in this Program, the developer retains all patent rights and that the total cost to manufacture, including tooling costs plus a reasonable profit, is provided to the developer.

The Incentive Program is interrelated with the Research and Development Program in that, where desirable, a partially successful candidate in the Incentive Program could receive research funding to further improve the engine system and this funding could come from the Research and Development Program.

Lastly, we mention that the Incentive Program is expected to provide a valuable source of information from actual vehicles from which to judge the capability of the industry to meet 1975/76 emission standards.

This Clean Car Incentive Program has just begun, with approximately 20 proposals from industry to enter the prototype phase received recently, and more to come. While proposal evaluation is not as yet complete, as we see it now, delivery of the first prototypes of low emission cars into this Program will be made before August of this year. Figure 17 summarizes Program status.

The Incentive Program has been planned at $20 million over a three year period. The rate of use and extent of use of these funds is dependent on the rate at which selected candidates proceed through the Program. At each test stage, any given candidate can be eliminated. The period of major costs should be in 1972 and 1973 since a number of candidates would be entering, or, in the demonstration test stage in 1972 and the fleet test of one or more candidates should begin in 1973.

PROGRAM STATUS

*MARCH 1971 CONTRACTS (approx. 15) FOR

- Brayton Cycle Gas Turbine
- Rankine Cycle
- Heat Engine/Electric Hybrid
- Electric

*PROTOTYPE DEMOSTRATIONS IN 1971 WITH GAS TURBINES

Figure 17.

World-wide Participation in Both Programs

Both Programs are open to firms residing outside of the United States. Most contractual work in the Research and Development Program is awarded based on competitive proposals by industry in response to Scopes of Work which are well publicized and then sent to firms requesting them. Multiple awards or contracts are common to the Research and Development Program. For example, we now have three different firms under contract and working on three different approaches to develop high efficiency combustors for the Rankine cycle system. There will be four parallel approaches considered for the gas turbine combustor design. As a step toward improving lines of communication between ourselves in the Program and industry in your country, we intend to define mechanisms to provide the Scopes of Work for new contracts to you and other interested industrial firms. In addition, the Scopes of Work for all on-going contracts, and final reports from completed contracts will be made available to all interested parties.

We welcome your participation in both Programs. Any firm or government representative who wishes information on participation will be provided written material in this meeting.

Summary

I will now summarize this talk. We have defined the Research and Development Program and the Incentive Program. Together, they form the Advanced Automotive Power Systems Program. Their goals are similar--to produce a virtually pollution free automobile--their methods differ however. We feel confident that ultimate goal will be achieved within the 5-year time frame. If the conventional internal combustion engine can be modified adequately by industry, so much for the better. But, it is these two Programs which will provide the insurance to our nation in the event that they cannot make it.

Lastly, we urge that industry in your countries compete in the Programs. We will do all that is reasonable to encourage this activity. In this way all of us will benefit, here and at home.

Chapter IV

THE POTENTIAL OF THE GAS-TURBINE VEHICLE IN ALLEVIATING AIR POLLUTION

by

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Introduction

Emissions from motor vehicles account for approximately 60 percent of the annual total air pollutants emitted in this country (see reference 1); the three major pollutants emitted by automobiles--unburned hydrocarbons (UHC), carbon monoxide (CO), and nitrogen oxides (NO_x)--comprise approximately 63, 93, and 46 percent, respectively, of the total of each of these pollutants. Consequently, there has been a great deal of interest in ways and means of reducing the emissions of Otto cycle internal combustion gasoline engines, and of finding inherently cleaner-burning alternatives to these engines.

It is generally believed that if the reciprocating internal combustion engine cannot be cleaned up sufficiently in a reasonably economic manner, the gas turbine is the most likely alternative. Its inherent emission characteristics are generally considered to be satisfactory (presently or potentially), and the primary issue is whether it is technically and economically feasible. Based on several well-advertised programs of the major automobile manufacturers in the United States and abroad, it appears that the technical problems are surmountable, but there is, as yet, considerable doubt concerning the economics of automobile gas turbines.[1]

The purpose of this paper is: (a) to summarize the emission characteristics of automobile gas turbines, (b) to examine the potential of these engines for satisfying automobile requirements, and (c) to assess their ability

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(1) A study sponsored by the National Air Pollution Control Administration is being conducted currently by the United Aircraft Research Laboratories, with the purpose of examining the probable manufacturing costs of several candidate automotive gas turbine engines, including those with simple and regenerated cycles, and with free and single-shaft turbines. The results, when published, should provide an interesting data source in an area where the available literature is scarce.

to compete with cleaned-up reciprocating internal combustion engines by identifying their relevant technical (performance) and economic (manufacturing-cost and fuel-consumption) characteristics. The paper is intended to provid background information for a panel discussion at the 1970 ASME Winter Annual Meeting.

Emissions

Uncontrolled Emissions of Otto Cycle-Engined Automobiles--An automobile utilizing an Otto cycle (gasoline fuel, reciprocating) engine with no emission control devices has three sources of emission--the exhaust, the crankcase, and the fuel supply system (fuel tank and carburetor). Virtually all of the CO, NO_x, and lead, and about 60 percent of the UHC come from the exhaust of the vehicle; about 30 percent of the UHC comes from the crankcase; and the remaining 10 percent of the UHC comes from the fuel supply system (see reference 2). In terms of quantity of emissions, the pollutants in the exhaust, based upon a weighted average of engine sizes, are 0.50 lb/hr UHC, 3.5 lb/hr CO, and 0.17 lb/hr NO_x (see reference 3). Variations of ± 20 percent or more are typical. The total emissions from an uncontrolled Otto cycle-engined automobile have been estimated as follows (see reference 1): unburned hydrocarbons--520 lb/yr; carbon monoxide--1,700 lb/hr; and nitrogen oxides--90 lb/yr.

Emission Standards--The first nation-wide standards for automotive emissions were issued in 1966, and were to take effect on all 1968 model year cars and light trucks (see reference 4). These standards are given in Table 1 in terms of concentration, and pertain to engines with 140 cubic inches or large displacement. The standards for 1970 model year cars, also shown in the following, tighten the 1968 standards by about 30 percent. An important change in the standards is a change from concentration (ppm) to a mass-per-mile basis. It will be noted that these standards do not contain restrictions on NO_x or evaporations.

Table 1

	1968 Model Year	1970 Model Year
Crankcase	Zero	Zero
Exhaust		
Unburned Hydrocarbon	275 ppm	2.2 g/mi
Carbon Monoxide	1.5% volume fraction	23 g/mi

Proposed future emission standards, in g/mi., are presented in Table 2, based on data in reference 5.

Table 2. Emission Standards

Pollutant	Model Year 1971	1975	1980
Unburned Hydrocarbon	2.7*	0.5	0.25
Carbon Monoxide	23	11	4.7
Nitrogen Oxides	--**	0.9	0.4
Particulates	--**	0.1	0.03
Total	21.8	12.5	5.4

*Includes evaporative losses from fuel supply system.
**No defined standard; estimated output for uncontrolled emission is 5.8 g/mi of nitrogen oxides and 0.3 g/mi of particulates.

The proposed 1971 model year standards include an evaporative loss, and the proposed 1975 model year standards include NO_x and particulate requirements. The 1975 standards represent an approximate 80 percent reduction in

UHC, and a 50 percent reduction in CO from the 1971 standards, and an estimated 85 percent reduction in NO_x from the uncontrolled automobile. The proposed 1980 standards are those put forth by the President's Environmental Advisory Board and represent reductions by about 50 percent in the 1975 standards.

Several methods of adapting the gasoline engine to meet these standards have been proposed, and major efforts are being undertaken by the automotive industry to meet them. However, the effectiveness of some of the control devices has been questioned (see reference 6), and, consequently, alternative propulsion systems have been proposed. There is considerable evidence to suggest that the gas turbine has the potential of low emissions without control systems.

Emissions from Gas Turbine Engines--A gas turbine does not have a component that serves a function similar to the crankcase of the Otto-cycle engine. Lubrication, where necessary, is usually by a method which could be referred to as dry sump, with no chance of the fuel-air mixture venting to the atmosphere from the oil tank.

The fuel supply system of a gas turbine consists of a fuel tank, injection pump, and injectors; hence, there are no float chambers or other reservoirs to allow evaporative losses such as those from the carburetor of automotive reciprocating engines. Since a turbine would probably operate on lower volatility fuels, such as kerosene or diesel oil mixed with low-octane gasoline, the evaporative losses from the fuel tank are more easily minimized.

Since neither the fuel system, the lubrication system, nor the fuel, itself, (in terms of lead or sulfur content) is likely to be a source of pollution from the vehicular gas turbine, it is evident that the combustion process is the only significant area of concern. As described in the following, the continuous combustion process at overall lean fuel-air ratios leads to inherently low emissions levels of UHC, NO_x, and CO for gas turbine engines.

The concentrations of exhaust pollutants in gas turbine cannot be compared readily with those of the Otto-cycle engine because of the disparity of fuel-air ratios. Therefore, a method of comparison using a parameter called the emission index (see reference 7) was devised. The emissions index (EI) is, essentially, a measure of the pounds of pollutants emitted per pound of fuel burned; it can be calculated by using the concentration of pollutant and the fuel-air ratio of the engine.

The overall lean fuel-air ratios and the absence of quenching on cool walls results in complete combustion and, consequently, low levels of UHC and CO emission. The UHC's are of two major types; the first is formed by those hydrocarbons that pass through the combustion chamber unchanged; the second consists of those which are produced in the low-temperature region of the combustion process. The formation of both types, as well as the formation of CO, takes place in the outer zones of the combustion chamber, where the air injected to cool the liner walls has lowered the local temperature. The emissions index is highest at idle for UHC and CO.

As the engine power output increases (i.e., the equivalence ratio increases), combustion improves, the emissions index for UHC and CO becomes negligible, and the absolute emission levels decrease significantly--despite the increase in fuel consumption due to higher power output.

As in the Otto cycle engine, the formation of NO_x is probably caused by the oxidation of the nitrogen in air $(O + N_2)$, and measurements to date have indicated that it is a strong function of temperature and a weak function of pressure, at least for practical gas turbine cycle pressures. These measurements show NO_x levels distinctly lower than those expected for equilibrium conditions (see reference 7), in contrast with the NO_x levels of Otto cycle engines. Furthermore, the NO_x is essentially frozen at the combustor exit, and concentrations remain the same during the expansion process. Among the reasons postulated for this low NO_x concentration are the rapid quenching of the combustion products with the excess air introduced into the combustor for cooling and the locally rich burning near the fuel droplets in the primary combustion zone. Since temperatures at the combustor exit are a function of both combustor inlet temperature and the equivalence ratio, it has been assumed commonly that improvements in cycle efficiency leading to higher combustor inlet temperatures (as a result of higher pressure ratios, or of regeneration) or higher turbine inlet temperatures will automatically lead to higher NO_x emission levels, because of the associated higher combustor temperatures. This assumption is not necessarily valid. Analytical and experimental research conducted at the United Aircraft Research Laboratories has indicated that the bulk of the NO_x formation occurs due to the strong recirculation patterns characteristic of the primary combustion zones, since in straight-through flow, the exposure of the mixture and products to high temperature in the flame front is not of sufficient duration to allow significant NO_x formation (see reference 8). Thus, the prospect exists that further research, which yields knowledge of how variations in primary combustion zone geometry, residence time, combustion temperature, and fuel-air ratio affect overall NO_x production, may indicate how combustor designs must be modified in order to reduce the already low NO_x emissions level of vehicular gas turbines even further.

Comparisons of Automobile Engine Types--Table 3 presents comparisons of the emissions index (ratio of pollutant weight to fuel weight x 10^3) for several automobile reciprocating engines and types of gas turbines, reported from various sources and compiled in reference 7. Although these data are not completely compatible and despite their wide variation, they can be used as a basis of some general conclusions concerning the emission of CO from a typical gas turbine. Using the EI as a basis, the emission of CO from a typical gas turbine is from 1 to 10 percent of an uncontrolled Otto-cycle engine, the emission of UHC is 1 to 20 percent, and that of NO_x from 30 to 80 percent.

Table 3. Comparison of Emission Indices

Engine/Engine Pollutant	CO	UHC*	NO_x
Reciprocating (30 mph)	241	6.1	16.3
Reciprocating (cold start)	513	35	14.5
Reciprocating	407	35	13
Regenerative Gas Turbine (30 mph)	5.3	0.3	13.5
Gas Turbine (cold start)	50	1.1	9.8
Aircraft Turbojet	3.3	0.4	5.5
Aircraft Turbojet	19	2	5

*Expressed as hexane.

The procedure of totaling emission indices, used in this paper, is adopted in order to provide gross comparisons between the two types of engines. It is felt that this simplified approach has validity, since the emission standards of Table 2 are based on reciprocating engine outputs, rather than specifically on precise health-based criteria for particular pollutants. On this EI basis, the gas turbine ranges from 1 to 11 percent of that of the untreated Otto-cycle engine.

Korth (see reference 9) and others have measured emissions from regenerated automobile gas turbines operating in typical duty cycles. It is very difficult to instrument mobile vehicles properly for accurate measurements of the extremely low concentrations of pollutants typical of gas turbine engines, and wide variations have been reported among several makes of engines. Therefore, extrapolations of the test-stand measurements of an aircraft gas turbine to the automobile standard duty cycle are of interest. Although such an engine would not be applied to an automobile, its emissions characteristics might closely parallel those of a simple-cycle automobile engine of acceptable efficiency. For such an engine applied to an automobile type duty cycle, an average total EI of 10.0 may be assumed which, at an assumed fuel consumption of 270 g/mi^{4}i^{1} yields a total emission (UHC + CO + NO_x) or 2.7 g/mi. Thus, the extrapolated emissions level is approximately 50 percent of the proposed 1980 standard for the treated gasoline internal combustion engine, even though its NO_x levels exceed the standard for that particular pollutant.

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(1) About 11 miles per gallon.

It must be emphasized that this emission level is predicted for gas turbine combustion technology of the 1960's, with no treatment, and with no emissions criteria considered in the combustor design. Furthermore, the ability of treated gasoline automobiles to continue to meet emissions criteria over the lifetime of the vehicle is open to serious question, even if costly maintenance and inspection procedures are adopted and enforced. In the absence of perfect maintenance, if a 1980 Otto cycle engine reverted to emissions of the level of the 1975 standard, it would cause as much pollution as 4.6 untreated gas turbines (as described in the foregoing); if it is reverted to 1971 standards, it would cause as much pollution as 11.4 gas turbines; and if it reverted to its natural untreated operating state, it would cause as much pollution as 35 gas turbines.

The occurrence of the poor mixture/excess blowby condition commonly observed on today's automobiles on the road would lead to even higher levels of pollution. For the gas turbine, poor adjustment of fuel to the combustor is unlikely to cause pollution problems. If the mixture becomes fuel-rich, the engine simply puts out more power; if the car were operating at maximum throttle, and the owner ignored overheat warnings, the engine might fail ultimately, but the pollution control of the gas turbine is fail-safe.

Alternative Power Plants--Other than the gas turbine, electric, steam, and Stirling engines are the most frequently mentioned of low-pollution alternative power plants to the gasoline reciprocating engine for vehicular propulsion. Well-reported efforts have been directed by various organizations toward investigating steam and electric alternatives to the gasoline engine. These efforts nearly always confirm the difficulties inherent with these approaches, lending support to the strong economic pressures to continue the usage of gasoline engines for automobiles, even though control of their pollution will involve substantial and costly modification. At present, the major automobile manufacturers are publicly committed to the reduction of emissions from gasoline engines as their preferred approach to the pollution problem, but have identified the gas turbine as their first alternative.

From an emissions standpoint, gas turbine emissions are likely to be roughly equivalent to those emitted by a central power station producing an equivalent amount of power for electrically powered vehicles.

Other Vehicular Applications--In terms of gross magnitude of emission of pollutants, the automobile far exceeds all other vehicles combined, both because of Otto cycle engine characteristics and sheer numbers. Despite the much lower levels of UHC, CO, and NO_x emissions contributed by diesel-powered trucks, buses, trains, and off-highway equipment of various types, special localized situations regarding pollution may favor the gas turbine for these applications as well. For example, the urban diesel bus, while theoretically a low emitter of pollutants, is severely criticized for introducing noise and smoke to sensitive areas, and gas turbines may alleviate these problems as well.

Conclusions Concerning Emissions--The technical literature documents the fact that the gas turbine, at least in most of its forms, does indeed have significantly lower emissions than the conventional Otto cycle engine and can meet proposed future standards for UHC and CO without modification (see references 7 and 9). The ability of the turbine to meet future NO_x standards is not as clear, although it is hoped that the NO_x emission will be amenable to solution when sufficient research and development are performed. Nonetheless, even without NO_x reduction, total untreated emissions (UHC, CO, NO_x) of the gas turbine are likely to be only 50 percent of those of the treated Otto cycle engine which meets proposed 1980 standards, and less than 3 percent of the output of an untreated Otto cycle engine. If all Otto cycle engines were replaced by gas turbines, the motor vehicle contribution to pollution referred to on page 1 would probably be reduced from 60 to 2 percent. In other words, it is likely that vehicular-caused air pollution would be eliminated as a cause for serious environmental concern.

Other Pertinent Characteristics

Engine Power Output--Figure 1 shows torque/speed characteristics of typical reciprocating engines and five gas turbines of various configurations, i.e., single-shaft, differential with 10 percent compressor speed, single-stage free-turbine, and multi-stage free-turbine configurations. Clearly, the wide variations in the characteristics imply that when discussing "typical" gas turbines, it is essential to specify the configuration. The configuration most commonly considered for vehicular applications(1) has been the single-stage free turbine, since the rising torque curve below rated speed yields positive stability when the output shaft speed is in a fixed ratio to the wheel speed. (As wheel speed drops due to increased resistance, the increased torque of the engine tends to restore the system to its original speed.)

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(1) The Chrysler, Rover, and General Motors automobile engines, and the GM, Ford, and Leyland truck engines have used the free-turbine configuration.

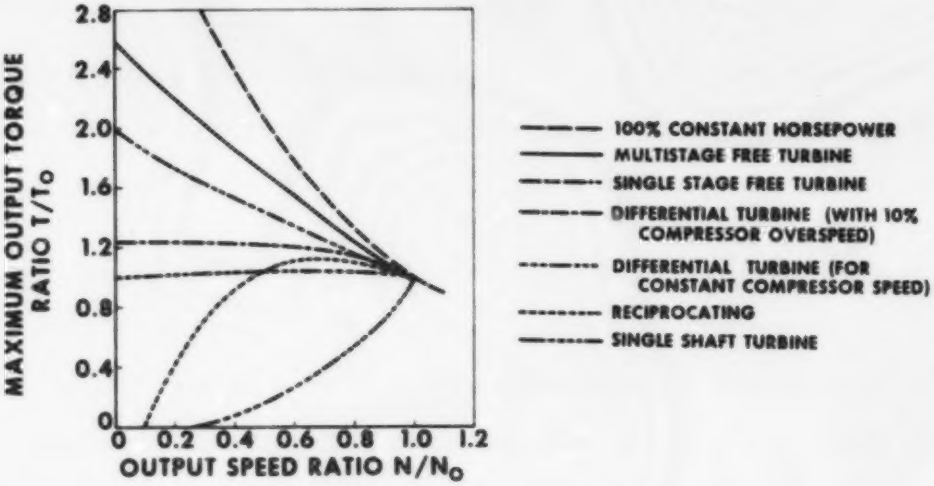

Figure 1. Engine torque speed characteristics.

The acceleration lag of gas turbine engines is due primarily to the fact that compression ratios are more constant in reciprocating engines (see reference 10). Nevertheless, many techniques are available to reduce the lag to acceptable limits. These include the momentary removal of load at some point in the power transmission train, reduction of engine shaft mass inertia, introduction of variable geometry, or allowing momentary overfueling (overheat) in a fashion analogous to the accelerator pump of an Otto cycle engine.

Because the gas turbine compresses far more air than is actually consumed in the combustion process, it is more sensitive to ambient conditions than its reciprocating counterparts: its power output and fuel consumption are both affected more adversely by high temperature and altitude, and the engine must be sized to provide adequate performance at reasonably adverse conditions. This fact implies that cars used extensively at high altitudes and temperatures should be equipped with higher powered engine options and also that certain potential "optional extras," such as water injection, might increase the flexibility of the engine.

Maintenance, Reliability, and Life--Experience with aircraft and industrial gas turbines has indicated that, because of their basic simplicity, these engines can operate for thousands of hours with only minor routine maintenance. The same can be expected for automobile gas turbines, provided their unique requirements are adequately satisfied in the design. One very significant requirement is the ability to withstand the thermal stresses associated with the frequent changes in power and, where regenerated engines are required, the addition of cores and seals may complicate the maintenance requirements.

Transmission Considerations--The engine torque/speed characteristics shown in Figure 1 are unsatisfactory for vehicle propulsion without torque multiplication. Examination of a spectrum of transmission configurations is needed to match the wide variety of torque/speed curves characteristic of various configurations of gas turbines to various vehicle requirements.

Ideally an infinitely variable (IV) transmission would be desirable for use with gas turbines. In effect, the IV transmission decouples vehicle wheel speed from engine speed and allows the engine to deliver full power at any wheel speed (subject to traction limitations and transmission efficiency), so that the torque speed characteristic of the engine is irrelevant to the output of the transmission. The lead candidates for vehicular gas turbine IV transmissions are electrical, hydrostatic, and mechanical.

The electrical system would consist of a generator (or alternator) driven by the gas turbine, an electric motor (or motors) driving the axle or wheels, and a control system. Although the electrical system would yield almost ideal power application, its manufacturing cost for automobiles is likely to be high. Hydrostatic transmissions, consisting of displacement hydrostatic pumps and motors, would require the development of successful turbine shaft speed pumps and might have noise and safety problems. Mechanical infinitely variable transmissions rely on rolling contact along a variable radius, and generally suffer from Hertzian stress problems at the contact surfaces. Therefore, they tend to result in comparatively large and heavy installations. Nevertheless, they merit serious consideration for automobile gas turbines.

The electrical transmissions described in the foregoing allow consideration of hybrid automobiles and buses (see reference 11) in which power is drawn not only from the prime mover, but also from energy storage devices, such as batteries of flywheels. These hybrid approaches permit shutdown of the prime mover (and thus elimination of all emissions) during operation in emissions-sensitive areas, and also permit charging the energy system at the most efficient and/or lowest emissions operating point of the engine, thus reducing overall emissions.

Transmissions with a finite number of gear ratios are attractive because of their low cost and high efficiency. For a given engine application, the higher the basic engine torque at reduced speed, the fewer the gear ratios needed; thus, a single-shaft engine would probably require an impractical number of gear ratios. Hydraulic torque converters are used in automobiles, where they are generally teamed with automatically shifting two- or three-speed geared transmissions; they are inexpensive, convenient, and adequately efficient. For a single-shaft gas turbine, such a transmission with a minimum of 5 or 6 gear ratios (or a multiple-stage torque converter) may furnish acceptable performance at reasonable cost.[1]

Vehicle Performance--The prime mover and transmission must be selected to meet some mission envelope of vehicle performance (desired accelerations, velocities, and elapsed times for a range of payloads). Standard computational procedures (see reference 12) relate propulsion power requirements, via transmission efficiencies, to road, grade, and air resistance; the acceleration performance depends, additionally, on the mass and rotational inertia of the vehicle. With appropriate allowance for installation and accessory losses, these procedures serve to establish the engine power requirements. Installed horsepower, as presently defined for automobiles,

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(1) For a free-turbine engine, the hydraulic torque converter is probably redundant.

can be misleading, since a typical automobile equipped with a nominal 250-hp engine will require about 30 hp to cruise continuously at 65 mph. (A truck at maximum gross weight equipped with a 250-hp engine, on the other hand, is likely to require its full power to cruise at that speed.) A typical automobile engine duty cycle, adapted from information supplied by Rover (see reference 12), is reproduced in Figure 2.

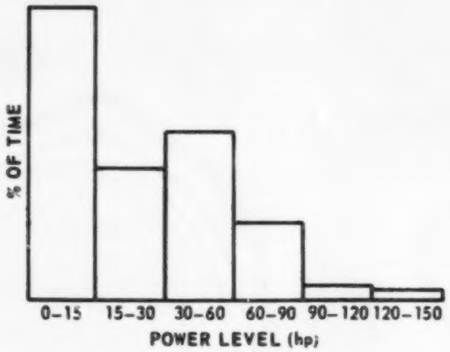

Figure 2. Typical automobile engine duty cycle.

The weight of the engine is important for establishing automobile performance. Present engines and radiators average 16.5 percent of total car weight (see reference 11), or approximately 660 pounds for the nominal 4,000-pound automobile. If the corresponding gas turbine weighed 160 pounds, the 500-pound saving would allow significant reductions in power output for a given performance level, since peak power requirements for the automobile are largely determined by the acceleration desired, and acceleration power is directly proportional to mass. For example, the power required at the road solely to accelerate a 3,500-pound automobile at 5 mph per second is obviously 12.5 percent less than that required for a 4,000-pound automobile (106.3 hp versus 121.5 hp at 50 mph). Applying standard road and aerodynamic resistance calculations to typical automobiles, summing power requirements, and assuming equal transmission efficiencies, leads to the conclusion that equivalent performance at 50 mph could probably be achieved with 11.8 percent less engine output power in the lighter vehicle (135 hp versus 153 hp).

Fuel Consumption

Historically, regenerators have been regarded universally as essential for practical gas turbines in automobile applications, because simple-cycle engines have been considered unsuitable for this application on the basis of poor full-load thermal efficiency and, even worse, part-load efficiency. Unfortunately, the addition of the regenerator considerably diminishes the appeal of the gas turbine engine in other respects, since it: (a) adds complexity, cost (manufacturing, maintenance and development), weight, and volume; (b) reduces specific power output and reliability; and (c) complicates control and response.

Simple-cycle engine efficiency (and specific power output) can be described conveniently by the use of only three parameters, i.e., pressure ratio, turbine inlet temperature, and flow path efficiency, where the latter represents both component efficiencies and pressure losses and is a measure of the aerodynamic sophistication of the engine (see reference 14). Figure 3 shows the relationship between full-load sfc and these parameters for a fixed flow path efficiency of 0.69 (see reference 15). As can be seen, a wide range of combinations of pressure ratios and turbine inlet temperatures will yield sfc's below 0.50, which has been the goal for regenerated engines for automobiles.

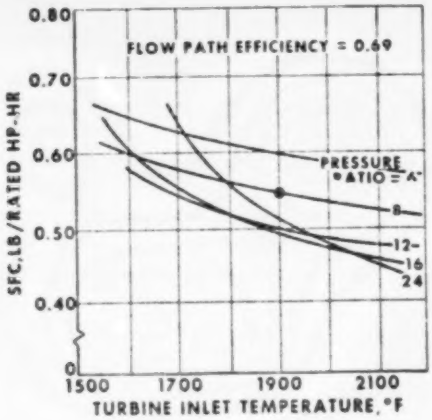

Figure 3. Fuel consumption relationship for simple-cycle engines with constant flow-path efficiency

The development of small, simple, advanced-technology components, with high efficiencies and high pressure ratios (see reference 16), permits reconsideration of the simple-cycle gas turbine as an automobile power plant. Although the fuel consumption of this engine will be higher than regenerative designs based on the same component technology,[1] and possibly slightly higher than that of competitive gasoline reciprocating engines, this engine may be competitive for several reasons: namely (a) automobile customers are not necessarily deterred by higher fuel consumptions (power mileages), provided the other features (including price) appeal to them; (b) the amount of fuel burned by the gasoline internal combustion

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(1) Both simple- and regenerative-cycle engine concepts based on small, simple, advanced-technology components are under study at United Aircraft Research Laboratories as part of the aforementioned manufacturing cost study.

engine, as well as the cost of the fuel may increase as a result of pollution control measures; (c) the gas turbine can burn lower-cost fuels; and (d) the possibility exists of governmental fuel taxation policy to favor the intorduction of low-pollution vehicles.

Manufacturing Cost--Although definitive production cost estimates are highly proprietary at this time, General Motors has released data indicating an opinion that the cost of automobile gas turbine engines will be approximately three times that of an untreated reciprocating engine, and from 1½ to 2 times that of an eventually pollution-treated reciprocating engine as shown in Figure 4 (see reference 17). These costs appear too high for serious competition with reciprocating engines, and, therefore, methods of reducing this level of manufacturing cost would appear mandatory in order for the gas turbine to replace reciprocating engines in quantities sufficiently large to have a significant effect on air pollution levels.

With regard to manufacturing cost, Eckert predicts that a proper gas turbine design adopted to modern manufacturing techniques should result in a cost division of 85 percent for materials and 15 percent for labor (see reference 10). Therefore, reduction of material costs are of primary interest in reducing manufacturing cost. Since superalloy material costs predominate in the raw material cost of today's gas turbine engine, one approach to reducing the manufacturing cost is to substitute ceramic materials wherever possible for refractory superalloys in order to take advantage of raw material costs of 5 to 40 cents per pound as opposed to $1.50 to to $6.00 per pound; however, the lack of tensile strength in ceramics in an impediment to this approach (see reference 18).

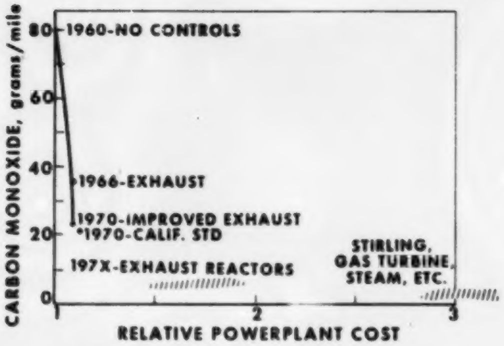

Figure 4. Carbon monoxide reduction economics.

Another approach is to increase the specific output of the engine by increasing pressure ratio, turbine inlet temperatures, and flow path efficiencies, thereby reducing the size and material content of cost-critical parts.

As stated in a previous footnote, a study is under way concerning the probable manufacturing cost of high specific output engines. Projections will be made concerning probable manufacturing costs of gas turbine engines for automobiles in units of 100,000 and 1,000,000 annually, based on material content and related costs.

Conclusions

Properly designed gas turbine engines for vehicular applications offer significant potential for alleviation of reciprocating engine-caused air pollution. From the emissions standpoint, this potential is represented by a predicted total emission level only 2 to 4 percent of that characteristic of untreated gasoline reciprocating engines and only 15 to 30 percent of that of reciprocating engines meeting proposed 1975 emissions standards.

Therefore, widespread adoption of vehicular gas turbine engines has the potential of practically eliminating vehicular air pollution as a subject of serious environmental concern, provided that the performance, efficiency, and manufacturing cost of these gas turbine engines are competitive with the reciprocating engines they would replace.

Acknowledgment

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Chapter V

THE STIRLING-CYCLE ENGINE

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Chapter V

THE STIRLING-CYCLE ENGINE

by

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Air Pollution is a very serious issue which is now causing considerable concern throughout the world. The convening of this meeting demonstrates this concern and we sincerely trust that it will not only highlight the problem but will also contribute to the solution of air pollution.

It is hoped that the intensive Stirling engine research programme in the Netherlands offers some positive contribution to the alleviation of the air pollution problem and in particular, the automotive pollution.

Based on our experience with this engine, its characteristics, the progressed state of the art and further prospects, we are convinced that the Stirling engine should be regarded as one of the promising candidates for alternative vehicle propulsion. It represents a very clean engine in the heavy duty vehicle application (on and off highway), and even for light duty vehicles.

The aim of this presentation is to provide an impression of:

(a) the nature of the work we have been doing and the resultant expertise which has been achieved
(b) the progress made
(c) the future potential of the Stirling engine and the envisaged applications
(d) the formulated plans for the future

It is impossible to adequately portray all the above facets within the time scale allocated for this presentation and for this reason tomorrow afternoon has been set aside to enable members of the conference to visit the facilities at Philips.

The initial development of the Stirling engine carried out over many years was not specifically directed towards vehicle use. However, this period was valuable in providing a sound basis for the later research and development phase which commenced during the early 60's.

Fig. 1 shows a 1-98 engine undergoing test. Engines are identified by the first digit indicating the number of cylinders and the remaining digits the piston displacement. About 30 of the 1-98 engines have been built used principally as test beds for various experiments. The original rating was 10 HP but later versions were rated at 20 HP. Maximum speed is 3,000 RPM.

Fig. 2 illustrates another engine concept namely an opposed piston type which can be used for ultra-silent underfloor boat propulsion. A 97% efficient wormgear reduction and coupling is built into the crankcase. Its rating is 100 - 200 HP at 3,000 RPM.

During 1968, it was decided to design a four cylinder in-line engine which could be used for vehicle as well as stationary applications, with a rating or 100 - 200 HP. Another criteria was that it should be capable of being mounted vertically or horizontally. Fig. 3 shows this engine.

The total experience on modern Stirling engines during the past ten years has been achieved on approximately 50 engines of seven different types. Powers range from a few tens of watts to 400 HP and the total running hours amount to 25,000. Component testing has also been a feature of the development programme and close to 1,000,000 hours have been logged over the ten year period. Engines have been endurance tested for 5,000 hours

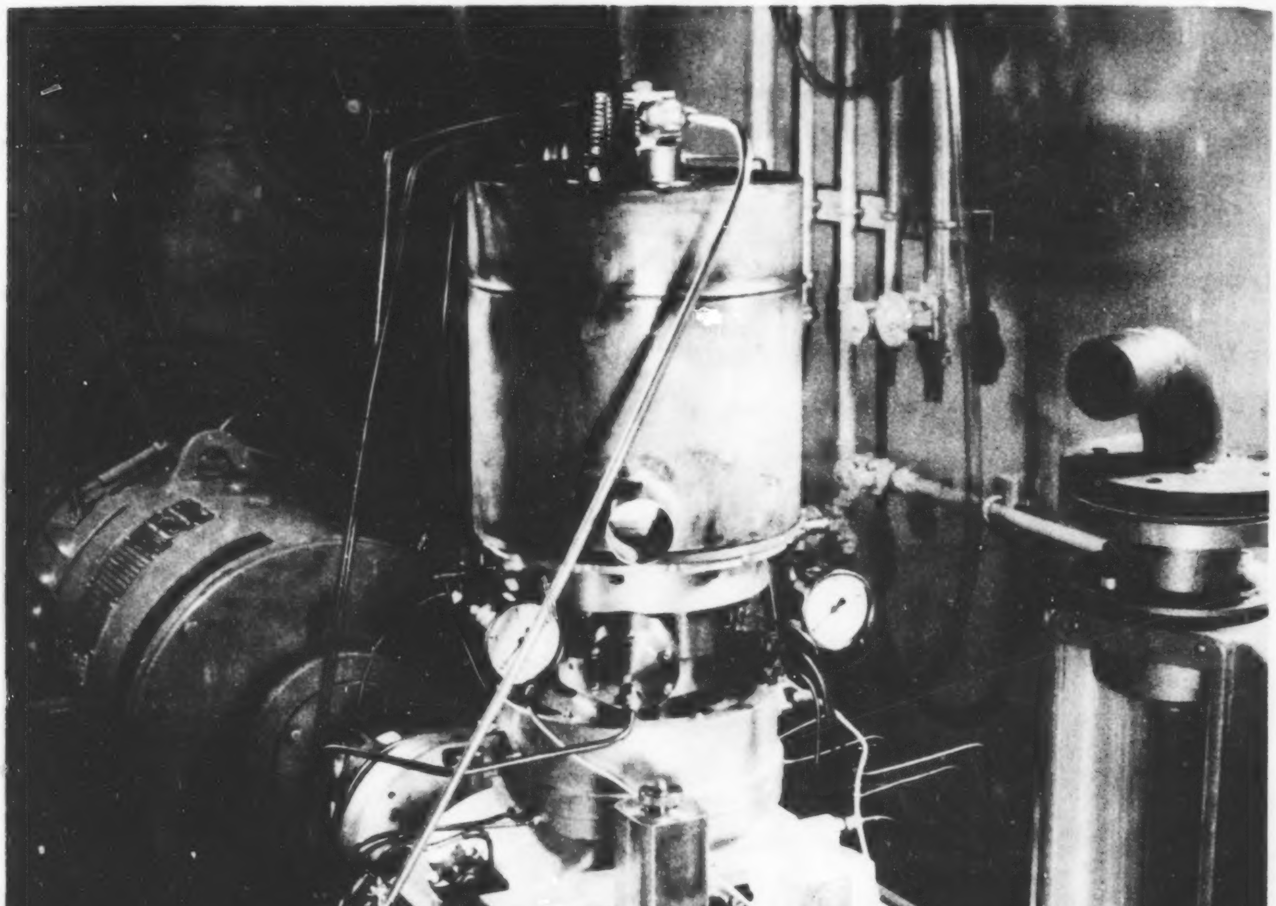

Figure 1. 1-98 Engine during test.

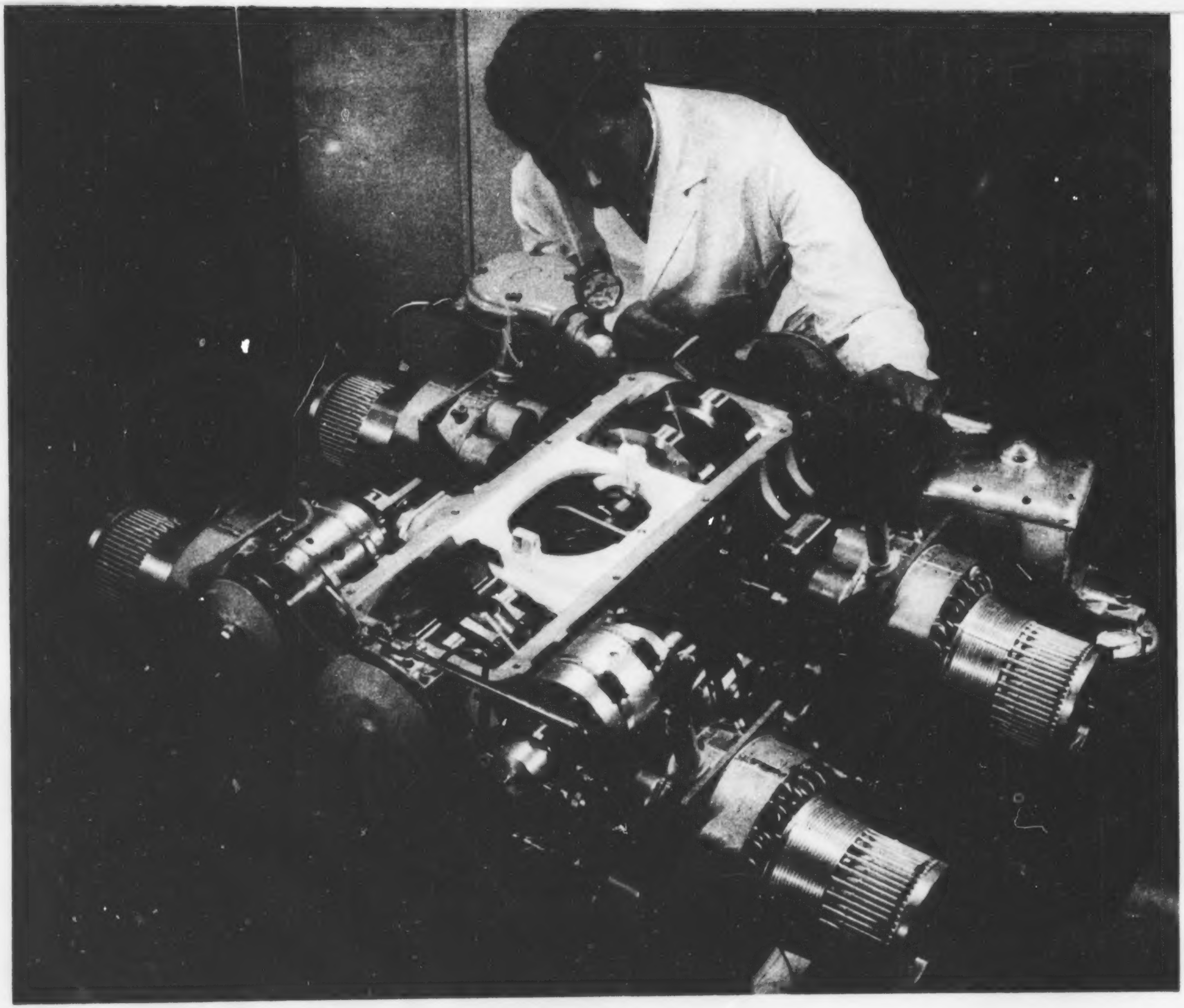

Figure 2. Four cylinder opposed piston engine.

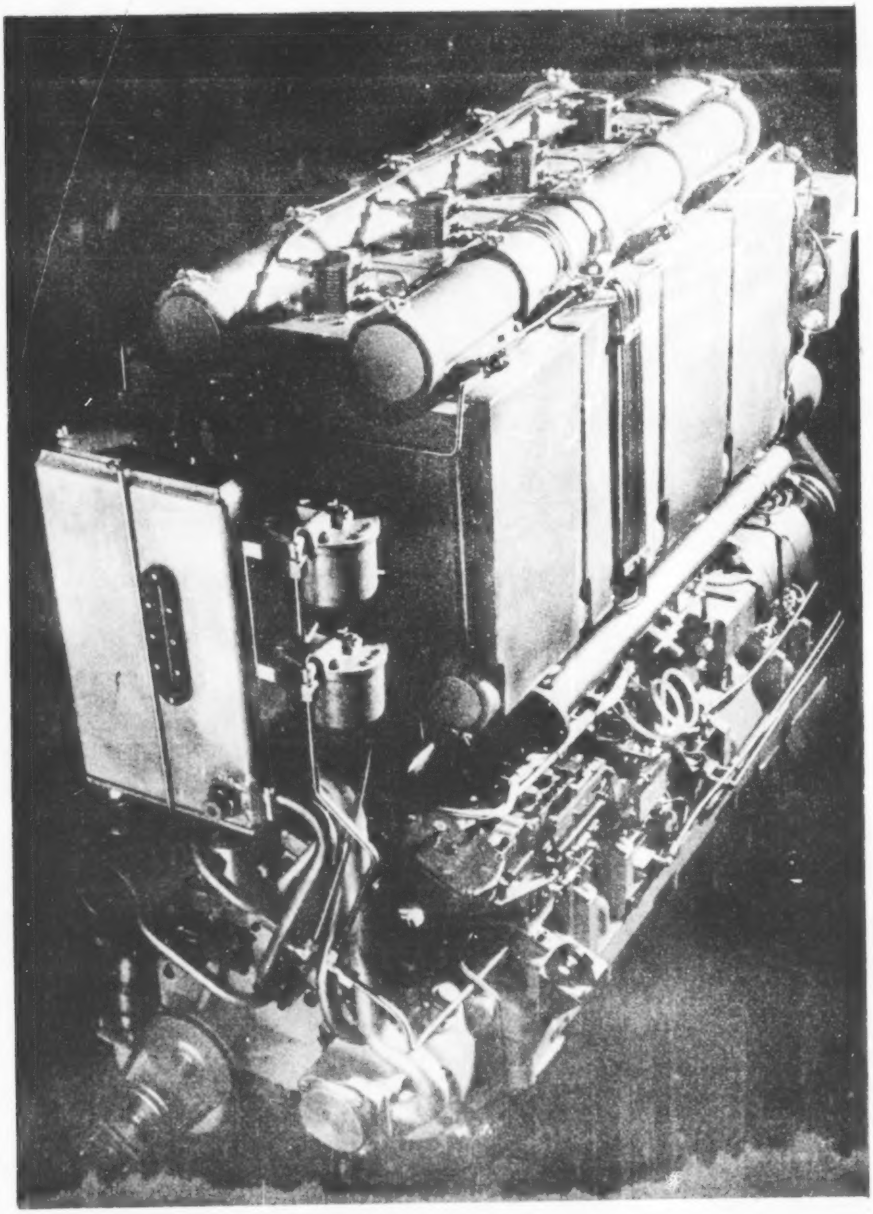

Figure 3. Philips 4-235 engine.

at full load and vital running components for over 10,000 hours.

We have a total of 30 dynamometer test stands. Fig. 4 shows three of them which are used to test the 4-235 engine.

During the past three years, a policy has been formulated which highlights the development of engines for vehicle applications as being of prime importance for these vehicle applications.

Emissions

Fig. 5 illustrates the external combustion system which is the basis of the modern Stirling engine. The power piston and the displacer are enclosed by the heater head, a composite unit comprising the burner, heater tubes, preheater and regenerator. The system has some excellent features.

The combustion is continuous and takes place in a space all surrounded by hot walls of 700°C (1,300°F), so there is complete absence of quenching effects, and combustion will be most complete. This is also accomplished by a very good mixing in the burner itself. Furthermore, the amount of excees air can be chosen freely, such that the design has a considerable measure of flexibility.

Due to the pre-heating of the incoming air the flame temperature is rather high; nevertheless, the amount of NO_x formed is very low. This is because local hot spots are avoided, and because the residence time in the burner is very short (5 to 10 milliseconds).

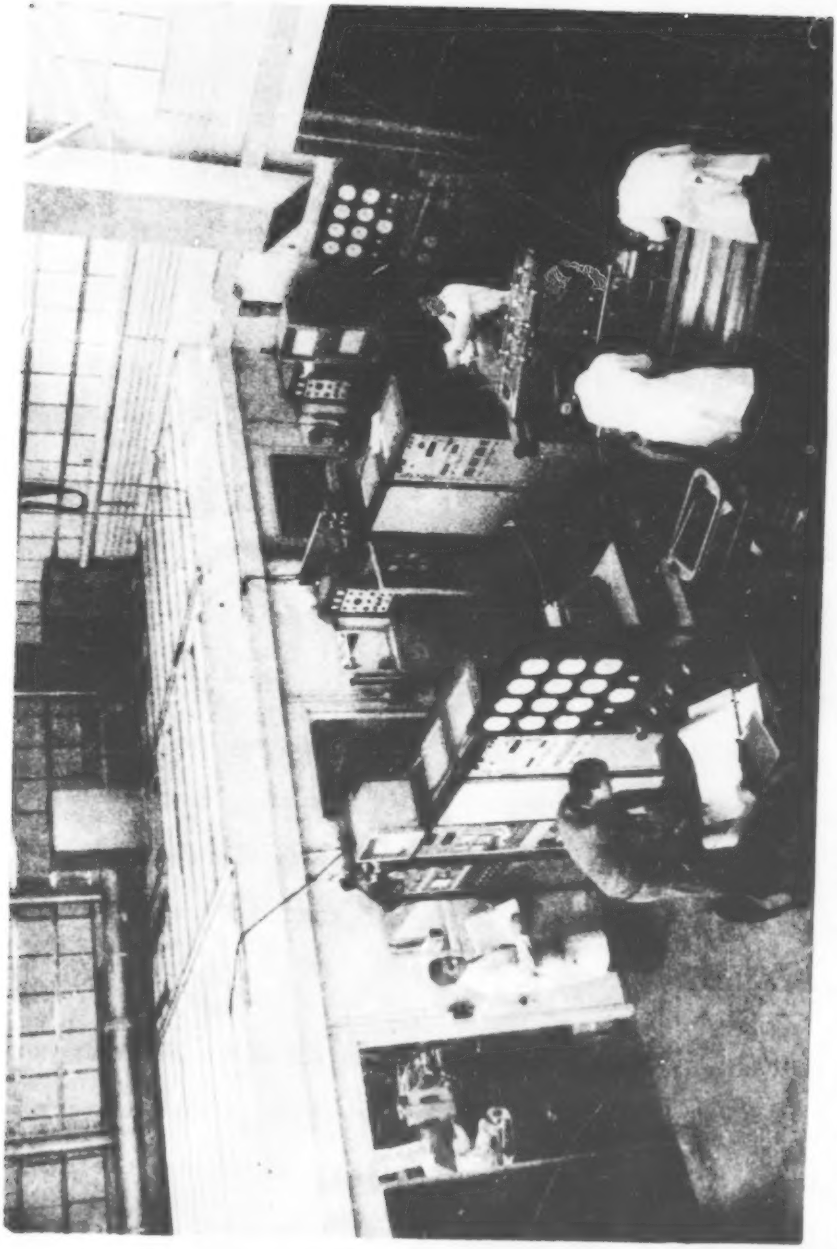

Figure 4. View of test cells with 4-235 engines installed.

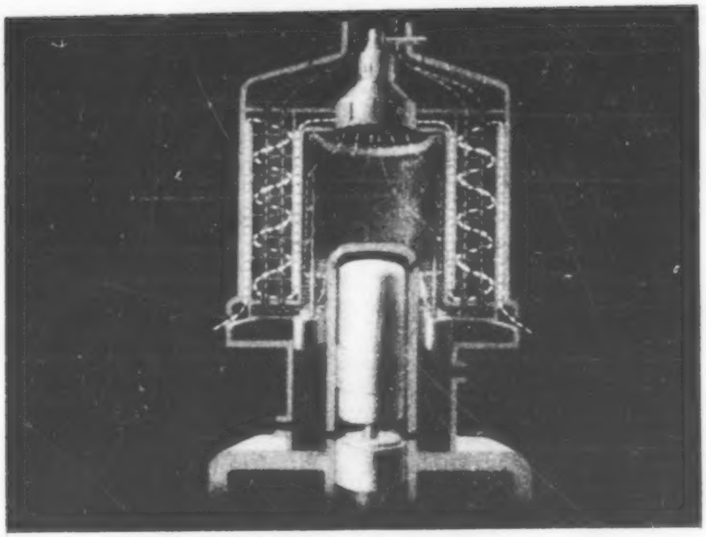

Figure 5. Heater head with piston and displacer.

STIRLING EXHAUST EMISSION (GVM)

	1968-71 CYCLE	1972-75 CYCLE	1970-71 STANDARDS	1975-76 STANDARDS	APSV PROGRAM
HC	.02	.03	2.2	.46	.14
CO	1.0	1.4	23.0	4.8	6.20
NO_2	.8	1.0	—	.4-.6	.40
30 % REC.	.16	.20			

Figure 6. Table of emissions.

Fig. 6 shows the results obtained on exhaust measurements. The emissions on a GVM basis are calculated for the old and the new cycle test. The standards and also the proposed standards are shown. It can be seen that C_x, H_y and CO are extremely low. The NO_x figures are valid for a burner which is not specifically optimized for low NO_x. New designs which use recirculation show measured values which lead to numbers as low as 0.2 GVM. Recirculation offers no particular problem because it only marginally effects engine output and efficiency.

It is also interesting to consider the comparison with the 13 mode Californian emission cycle for heavy duty vehicles which is shown in Fig. 7.

Other developments envisaged are for so-called suppressed flame temperature burners which also result in extremely low NO_x emissions.

Noise and Vibrations

The comparison between a Diesel and Stirling engine of the same power is shown in Fig. 8 and amounts to about 20 dB. The noise level from a Stirling engine is almost inaudible at a distance of 50 metres.

Controls

Fuel control is by means of a thermostat system. Power/torque is controlled by means of working gas pressure. A full load change in either direction can be made within 0.3 seconds. A cold start takes less than 20 seconds.

HEAVY DUTY VEHICLES (>6000 LBS) Gm/BHP. hr

	Stirling		1973 – 74 Calif.	1975 Federal
HC	.04	2.1 $(0.74)_{rec}$	12.5	5
NO_x	2.0 $(0.7)_{rec}$			
CO		2.2	40	25

Figure 7. Comparison with 13 mode California Emission Cycle.

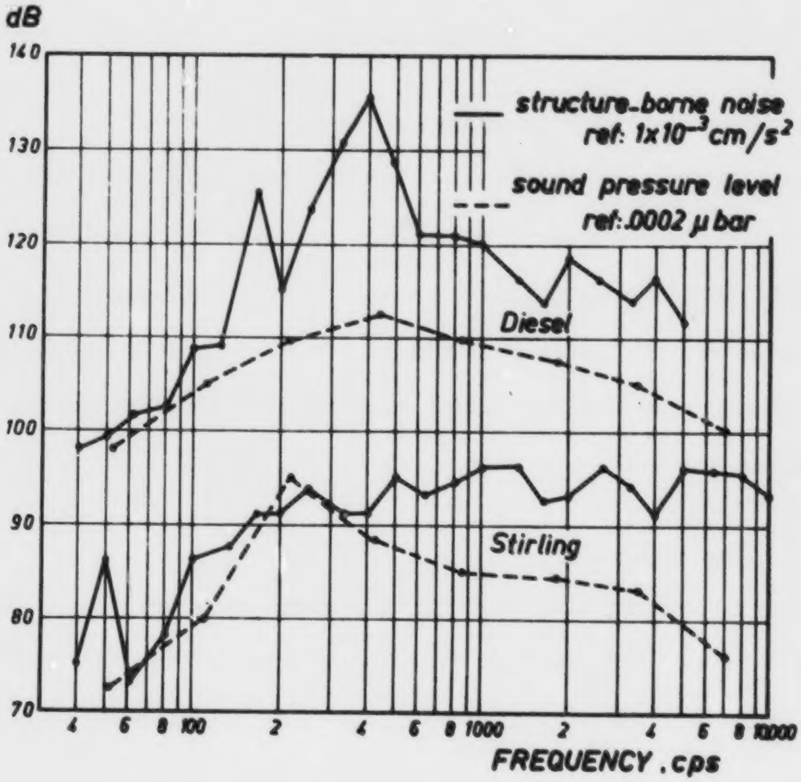

Figure 8. Noise level comparison.

Braking

By varying the gas pressure in the working and buffer space it is possible to obtain a braking effect up to 80% of full torque.

Overload

Engines are rated for continuous loads; short time overloading is possible.

Multi-fuel

The single cylinder demonstration engine in Fig. 9 illustrates the various fuels which can be used with a Stirling engine.

B.H.P./Efficiency/Torque

These characteristics are shown in Fig. 10 and 11 for various values of working gas pressure. Note the advantageous torque increase with decreasing speed compared with the Diesel engine.

The decision to concentrate on vehicle applications automatically poses the question as to what precise vehicle applications are envisaged. There is a distinction between professional or heavy duty vehicle propulsion and light duty vehicle propulsion (passenger cars). Different requirements as to power-weight ratio and initial costs in relation to fuel economy and maintenance and durability will hold. Stirling engines are (will be) suited for both categories.

Heavy Duty Vehicle Application

The 4-235 engine shown in Fig. 12 will be mounted in a bus and is mounted horizontally. It is initially rated at 100 HP with a capability of being increased to 200 HP. The weight is 750 K.g. The illustration shows the

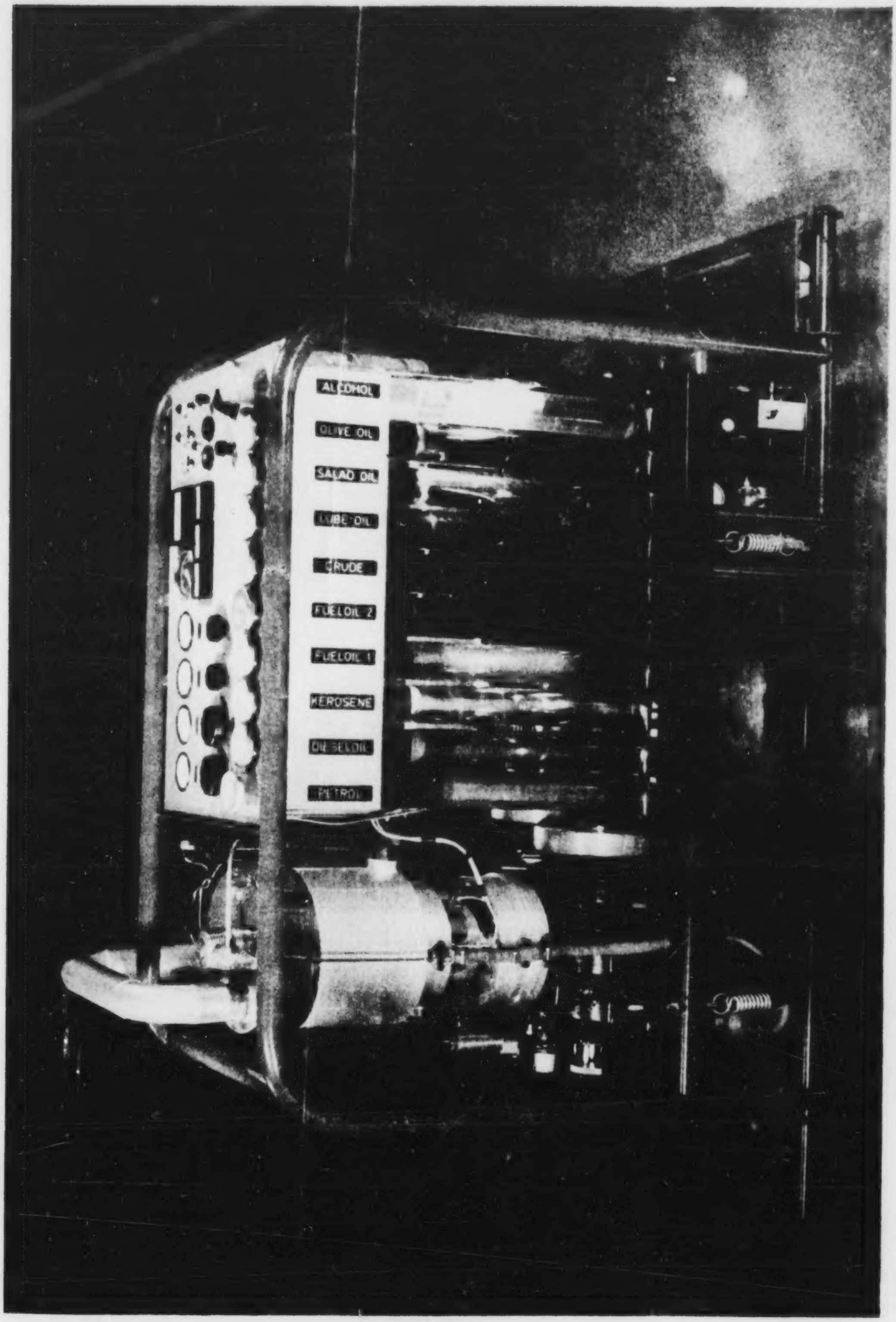

Figure 9. Multi-fuel demonstration engine.

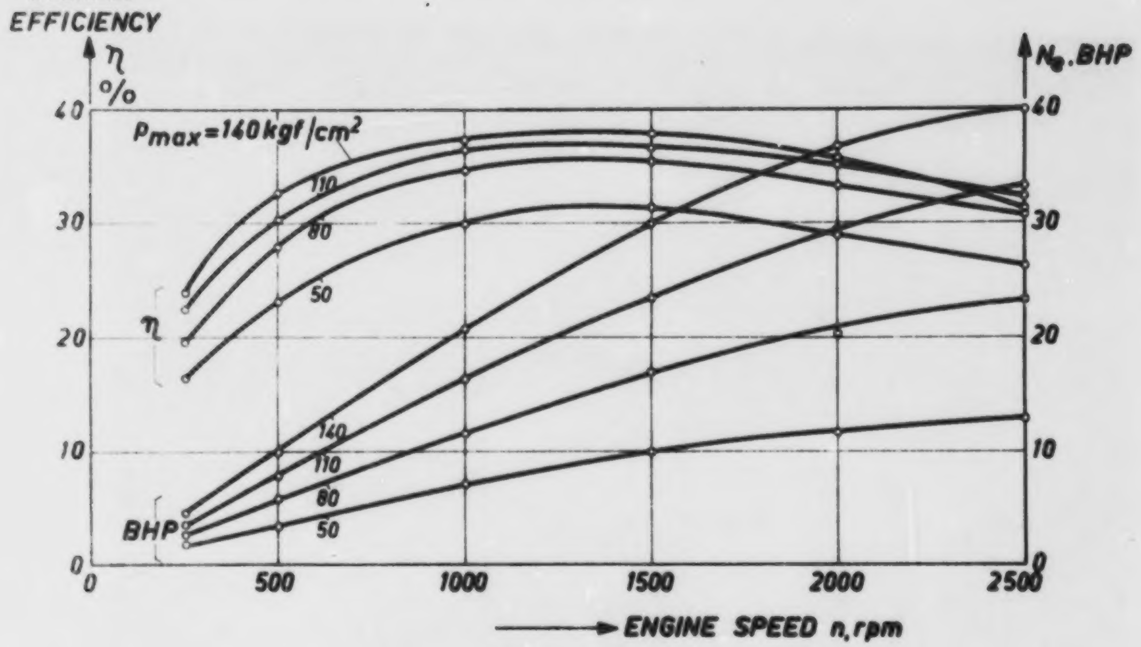

Figure 10. Efficiency curves.

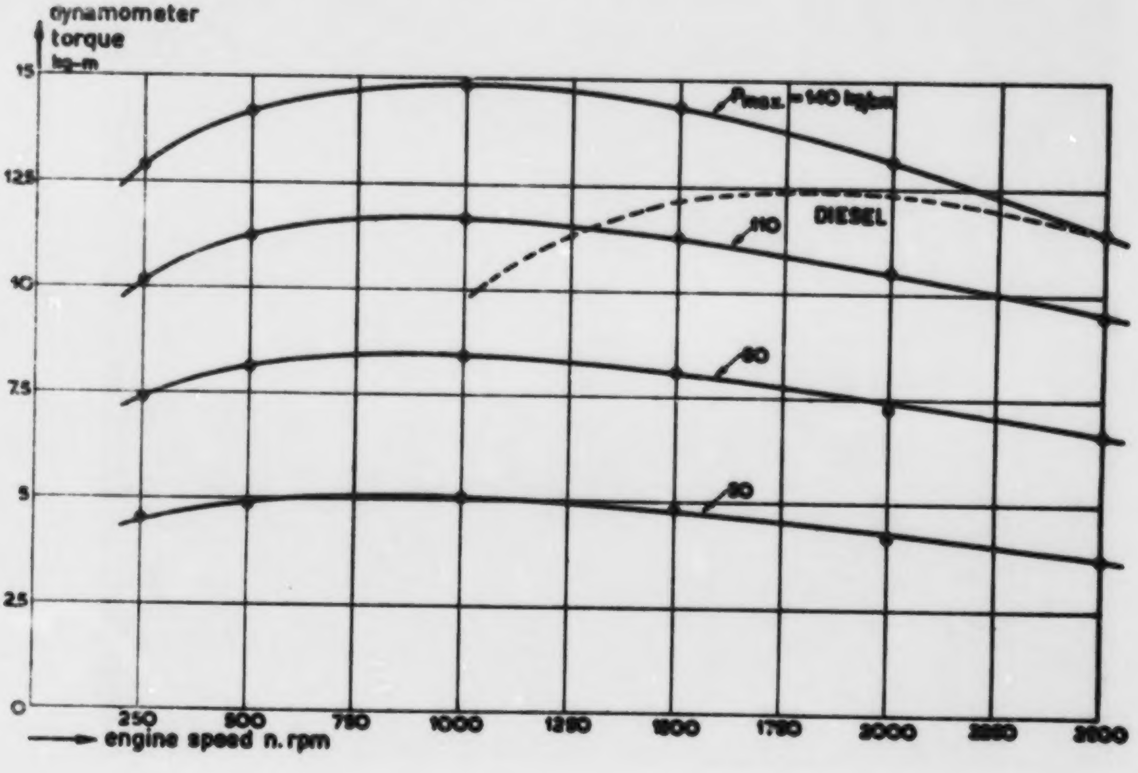

Figure 11. Torque curves.

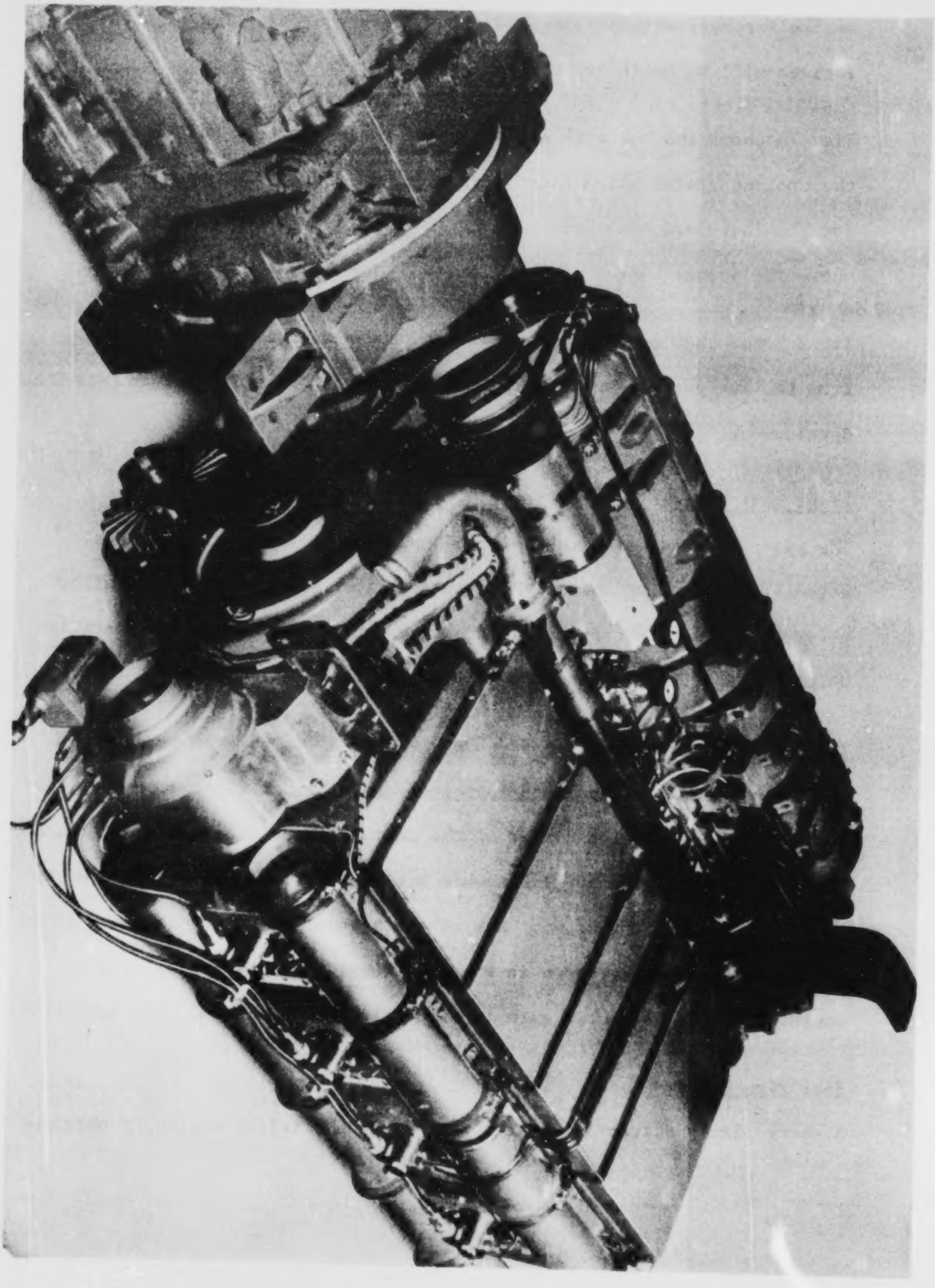

Figure 12. Philips 4-235 engine with gearbox.

engine assembly with a Voith-DIWA type 502-3 gearbox. Ten of these engines will be built for this and other applications.

Fig. 13 shows the bus with engine installed in the rear and also showing the cooling system which necessitates an additional radiator to cater for the increased heat load present in the Stirling cycle. Fig. 14 shows the actual bus.

It is hoped that a considerable amount of experience will be obtained from the bus project which will provide a sound basis for future development.

Light Duty Vehicle Application

The passenger car is undoubtably the application which commands most attention and one of the main characteristics required is a low weight power ratio. A different and attractive concept appears to be possible which would bring the ratio down to as low as 2 - 3 pounds per HP.

The double acting principle is shown in Fig. 15. The arrangement is that the hot space (expansion space) of one cylinder is connected to the cold space (compression space) of the next cylinder with a heater, regenerator and cooler between. The double acting principle can be achieved by more than one type of drive. One of the most promising of designs is the swashplate engine shown in Fig. 16. Besides a volume and weight advantage there is also a gain in simplicity of design and cost reduction.

Heat Pipe System

A particularly attractive heat source for the Stirling engine is obtained

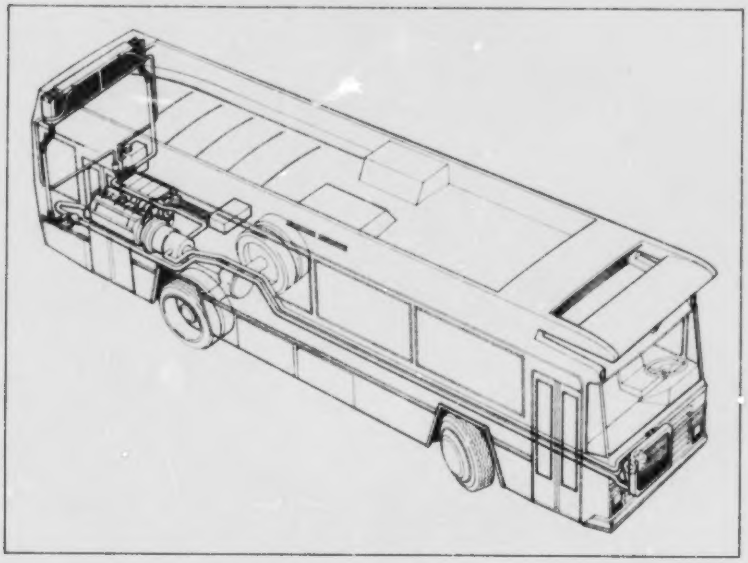

Figure 13. Installation of bus engine and cooling system.

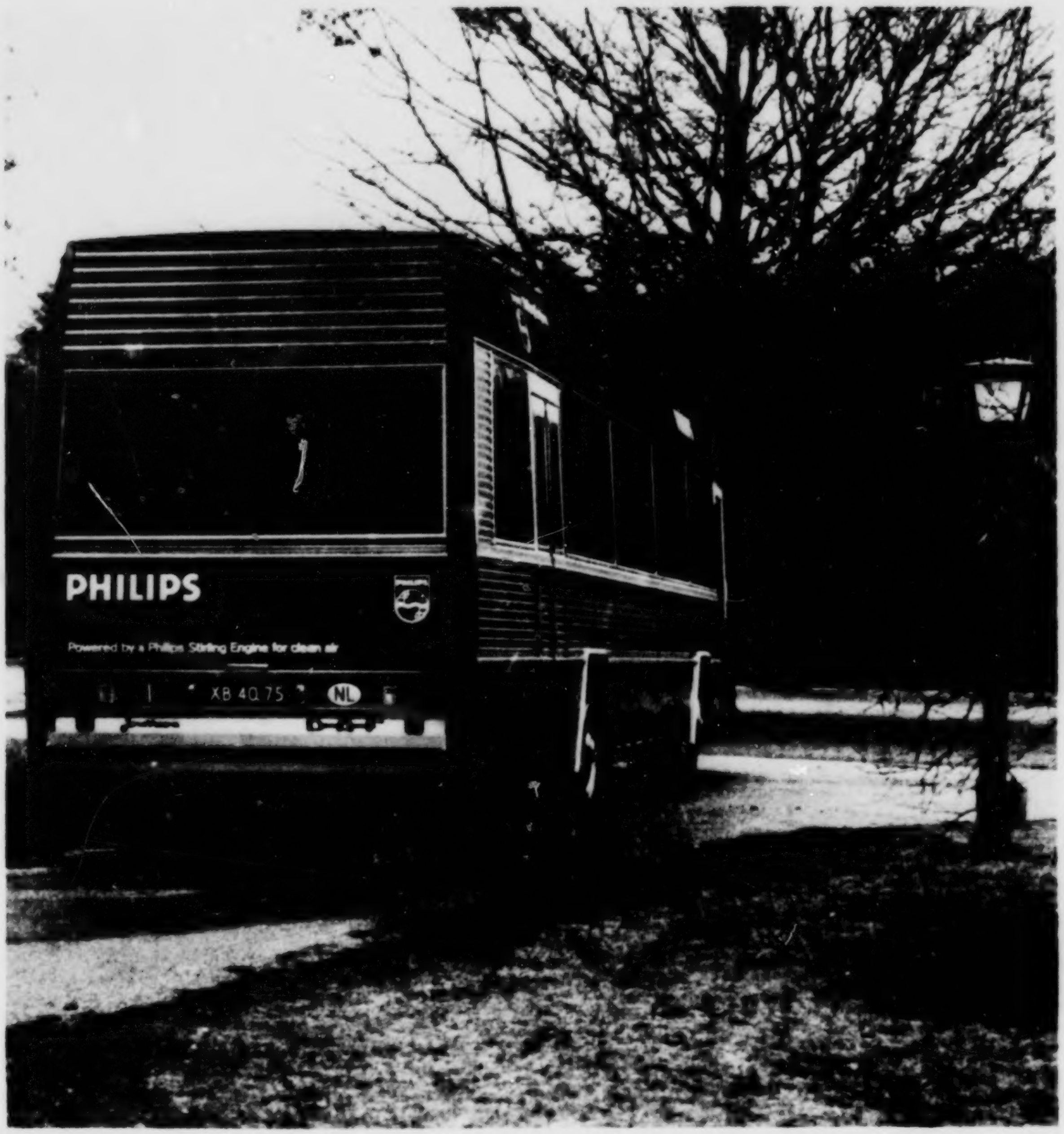

Figure 14. Bus powered by Philips 4-235 Stirling Engine.

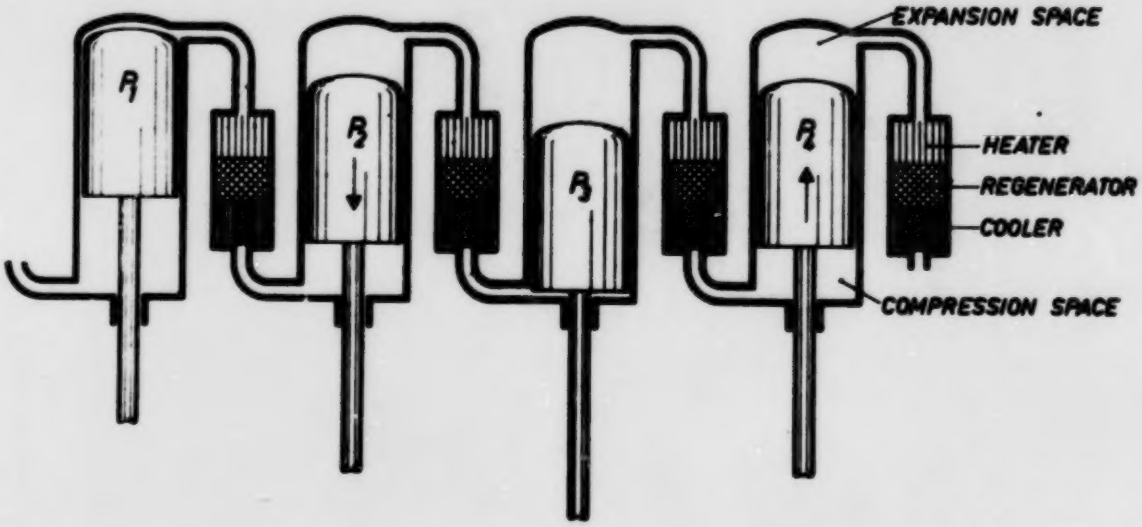

Figure 15. Double acting Stirling cycle.

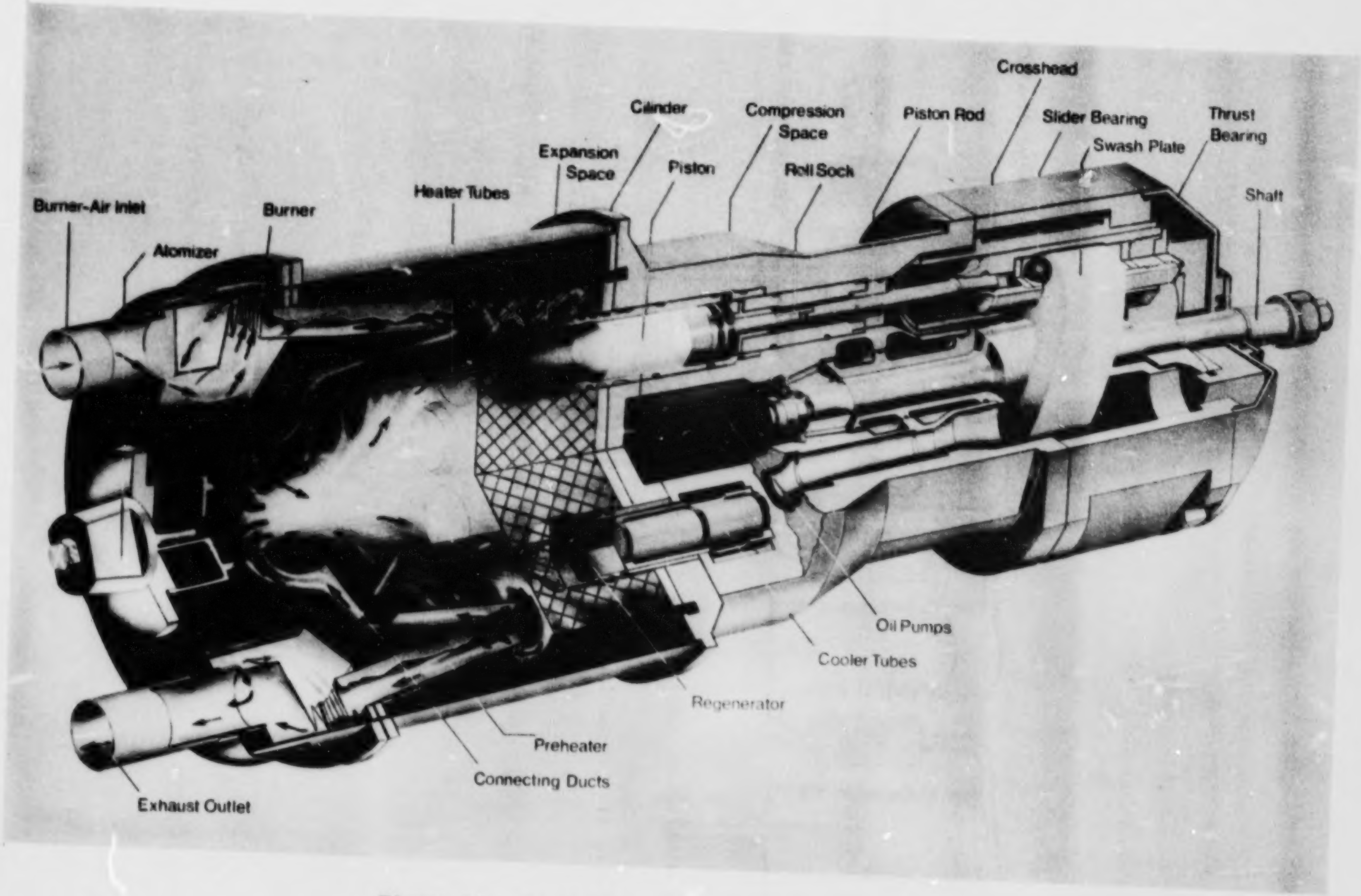

Figure 16. Swashplate engine.

by the application of heat pipes. A heat pipe is a closed volume in which is incorporated a medium which can absorb and reject heat at a very high rate; such a medium is sodium. By a process of evaporation and condensing, the external heat system of the Stirling engine is vastly improved. Fig. 17 shows a heat pipe system incorporated in a swash plate engine. Two major advantages emanate from such a design.

(a) External and internal heat transfer can be considered independently and the Stirling cycle can be optimized to a still higher specific output. It is possible to use higher speeds because heater flow losses are reduced.
(b) It is hoped that the combination with the combustor can be made such that the combustion process and heat transfer are partly simultaneous. The maximum flame temperature can be reduced to 1,600°C where hardly any NO_x formation occurs.

The development phase for such a promising combustion design has already commenced. Engines utilizing the heat pipe system represent engines which are commonly referred to as being of the "second generation."

Reviewing the proceeding engine types one endeavors to be realistic. The hardware experience gained has been considerable; soon we will acquire vehicle experience. However, the engine development stage is only just beyond the laboratory concept. A considerable development task particularly in the field of cost reduction and engineering lies ahead before an economical acceptable engine is produced. A considerable effort is also being made in the field of new fabrication methods. This effort in advancing Stirling engine technology is almost entirely carried out by Philips and its licensees in Germany and Sweden. Other license agreements are being negotiated.

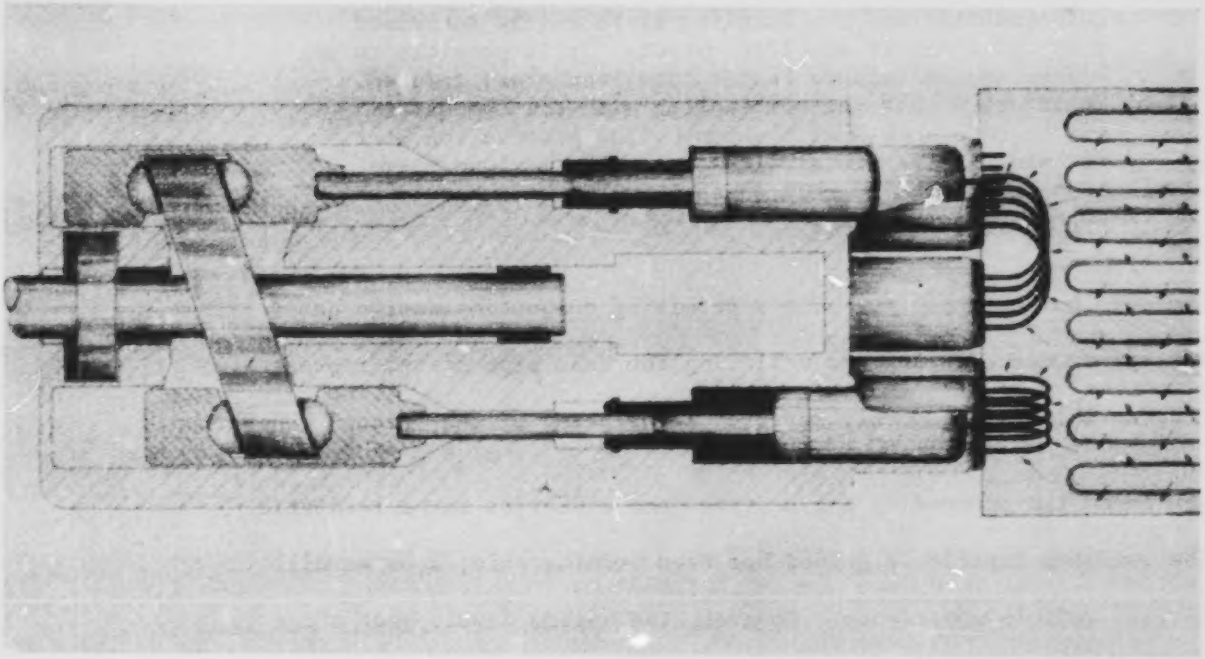

Figure 17. Swashplate engine with heat pipe system.

In Eindhoven the Stirling effort is divided into two main groups: the Research Group Stirling Engines and the Product Group Stirling Engines. The Research Group is primarily engaged in basic Stirling engine research while the Product Group is engaged in engine development for both vehicle and stationary applications.

Chapter VI

RANKINE-CYCLE POWER SYSTEM WITH ORGANIC-BASED FLUID AND RECIPROCATING EXPANDER FOR LOW EMISSION AUTOMOTIVE PROPULSION

by

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1. BACKGROUND ON WORK LEADING TO AUTOMOTIVE SYSTEM DESIGN

Thermo Electron Corporation has been involved in Rankine-cycle system development since 1963; this effort has been concentrated on completely self-contained portable powerplants in the horsepower range of fraction to ~ 200 hp. During this period the system has evolved from one of very limited practical application to one capable of competing with internal combustion engines in a number of commercial applications.

The initial work on Rankine-cycle systems at TECO from 1963 to 1967 was concentrated on steam as a working fluid. It was realized that steam had some severe limitations for a low-cost, reliable system capable of use at low ambient temperatures, but the development of alternative working fluids had not advanced sufficiently for practical consideration of their use. Since 1964, many new working fluid candidates have been proposed, and sufficient thermodynamic and physical property data have been developed for evaluating their potential as working fluids for commercially oriented applications. After extensive analysis of all of these fluids, thiophene was selected in 1967 as the current state-of-the-art working fluid which best fulfilled the system characteristics required for commercial Rankine-cycle systems. Development of components and systems using this fluid continued until September 1970, culminating in operation of a complete, self-contained 5 hp package.

The primary limitation for the use of thiophene in systems to be operated by the general public is its high flammability and toxicity. In September 1969, TECO began laboratory testing of trifluoroethanol, either pure or mixed with water, as a working fluid. The mixture of 85 mol% trifluoroethanol, 15 mol% water (Fluorinol-85)[*] was selected as the best composition. Fluorinol-85 has thermodynamic and physical properties very similar to thiophene and, in addition, is completely acceptable from a flammability and toxicity viewpoint for systems to be operated by the general public. Since September 1970, testing of a complete, self-contained 5 hp powerplant (see Figure 1) with Fluorinol-85 working fluid has been carried out with satisfactory results. Both thiophene and Fluorinol-85 have been operated in the same 5 hp package (after careful flushing and cleaning when changing fluids) with practically identical power and efficiency.

Fluorinol-85, with its completely acceptable safety characteristics, represents a significant advancement in Rankine-cycle technology for automotive propulsion systems. Work on use of the TECO Rankine-cycle system as an automotive propulsion powerplant was initiated in June 1969 under contract to the Division of Motor Vehicle Research and Development, Air Pollution Control Office, Environmental Protection Agency. During the first year of the program, a conceptual design of the complete system was developed. This conceptual design, described in Sections 2 and 3 of this report, indicated a strong potential of the TECO approach for a practical and competitive low-emission automotive propulsion system. The program is now in the second year and involves conversion of the conceptual design to a more detailed, optimized design and experimental development on several components, specifically:

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[*] Halocarbon Products Corporation, Hackensack, N. J.

Figure 1. 3 kw Generator Set with Side Panels Removed.

a. Analysis and bench-testing of full-size expander intake and exhaust valving approaches.

b. Detailed design, fabrication, and loop testing of a full-size feedpump.

c. Fabrication of a boiler third stage section and measurement of pressure drops and heat transfer rates.

d. Full-size rotary shaft seal testing.

In the remainder of this report, a summary description of the component designs, system performance and characteristics, and system packaging in an intermediate size American car are presented, based on the APCO-sponsored studies. The component designs are based on thiophene, since the details with Fluorinal-85 have not been completed. The differences are generally small; where differences occur with the Fluorinol-85, they are pointed out.

The system component sizes are based on the peak power requirements for an intermediate size American car to give a 0-60 mph acceleration time of ~ 15 seconds or better and a top speed of 90-95 mph Using these criteria, the system peak shaft horsepower is 100 shp. The reference car size for the APCO alternative powerplant studies was recently increased to a full-size American car. This modification will require an increase in the system power output of approximately 20% to meet the performance goals.

2. COMPONENT DESCRIPTIONS

2.1 System Working Fluid and Cycle Characteristics

The working fluid to be used in the system is Fluorinol-85, a mixture of trifluorethanol and water containing 85 mol% trifluoroethanol and 15 mol% water. As discussed above, sufficient experience has been obtained with this fluid so it can be considered a state-of-the-art working fluid. In Table 1, a summary is given comparing important cycle parameters for different trifluoroethanol-water mixtures; the equivalent calculation is also presented for thiophene, since this working fluid was used as the basis for the component designs given in this report. Based on these numbers, Fluorinol-85 was selected as the optimum composition, and is being used in testing the 5 hp system. This composition also has the minimum freezing point of -82°F; in Figure 2, the effect of water content on the freezing point is illustrated. The boiler outlet temperature and pressure are selected as 550°F and 700 psia, respectively, to maximize efficiency and minimize the engine displacement for a given power output. This cycle is illustrated on the pressure-enthalpy diagram of Figure 3. It should be noted that trifluoroethanol appears to be more stable thermally than thiophene, so that it may be possible to increase the boiler outlet temperature to 600°F during the program with a resulting improvement in cycle efficiency. While Fluorinol-85 appears the optimum composition at the present, future testing may indicate an alternate composition is optimum.

In Table 2, the overall design point cycle characteristics for the system are presented. The design point is based on a peak power requirement of 100 horsepower at an engine speed of 2400 rpm. For comparison, the design point characteristics are presented in Table 3 for thiophene, the working fluid used in the conceptual design study. The component designs presented in Section 2 as well as the overall system performance predictions of Section 3 are based on thiophene,

TABLE 1

CALCULATED CYCLE PARAMETERS FOR SEVERAL FLUORINOLS AND COMPARISON WITH THIOPHENE

Release Pressure = 75 psia
Condensing Temperature = 200°F
Subcooling T = Δ 20°F

Expander Thermal Efficiency, $\eta_{Exp_{th}}$ = 0.8

Expander Mechanical Efficiency, η_{Exp_M} = 0.9

Regenerator Effectiveness, η_{Reg} = 0.9

Pump Overall Efficiency, $\eta_{P_{OA}}$ = 0.7

Fluid	Boiler Outlet Temperature, °F	Boiler Outlet Pressure, psia	η Cycle, %	QR/QB*	Expander CID, in^3/hp at 2000 rpm	W_E, Shaft less W_P, Shaft Btu/lb
Thiophene	550	500	15.7	0.21	1.70	32.9
Fluorinol 100	550	400	14.2	0.31	1.84	26.5
		500	15.2	0.29	1.71	28.6
		600	16.0	0.26	1.58	30.4
Fluorinol 85	550	400	15.8	0.24	1.52	34.8
		500	16.2	0.20	1.40	36.3
		600	16.4	0.19	1.37	37.0
		700	16.6	0.175	1.34	37.2
	600	700	18.5	0.22	1.23	43.1
Fluorinol 61	550	500	15.4	0.14	1.56	44.4
		600	15.5	0.13	1.50	44.8
		700	15.9	0.11	1.42	45.9
		800	15.8	0.10	1.38	45.6
	600	700	17.0	0.16	1.42	50.2
Fluorinol 51	550	500	15.2	0.10	1.47	51.2
		600	15.7	0.087	1.40	52.9
		700	15.7	0.075	1.31	52.9
		800	15.6	0.061	1.26	53.3
	600	800	17.0	0.11	1.25	58.7

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*Regenerator heat transfer rate/Boiler heat transfer rate.

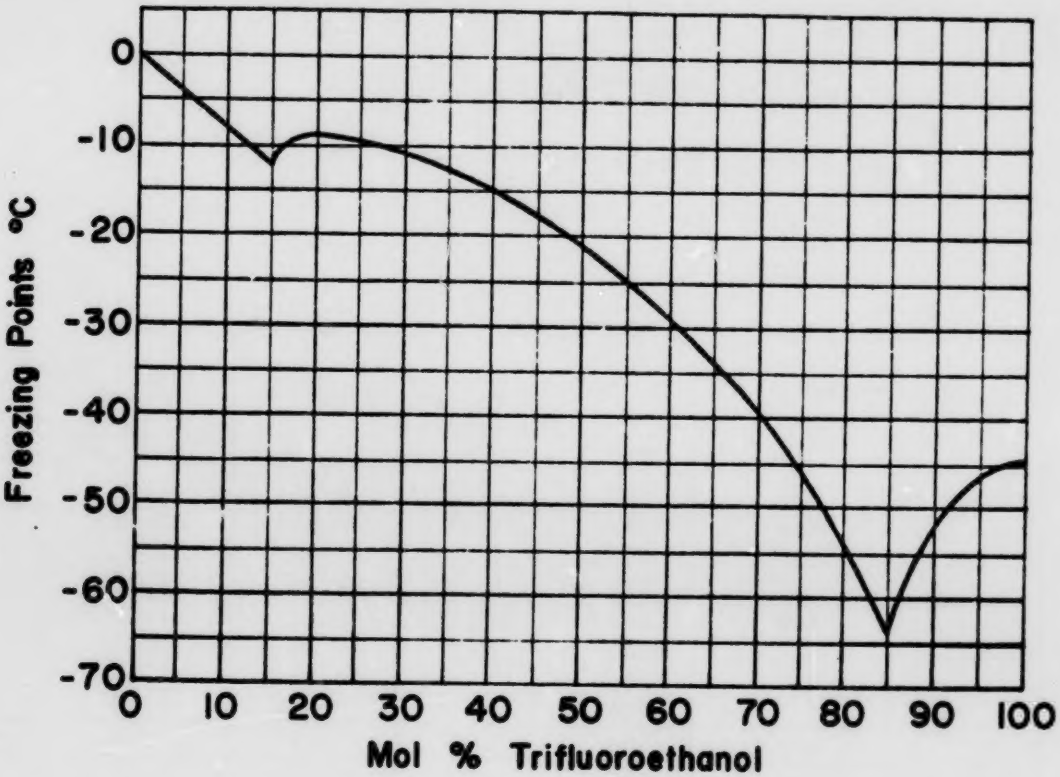


Figure 2. Freezing Point versus Water Content in Trifluoroethanol.

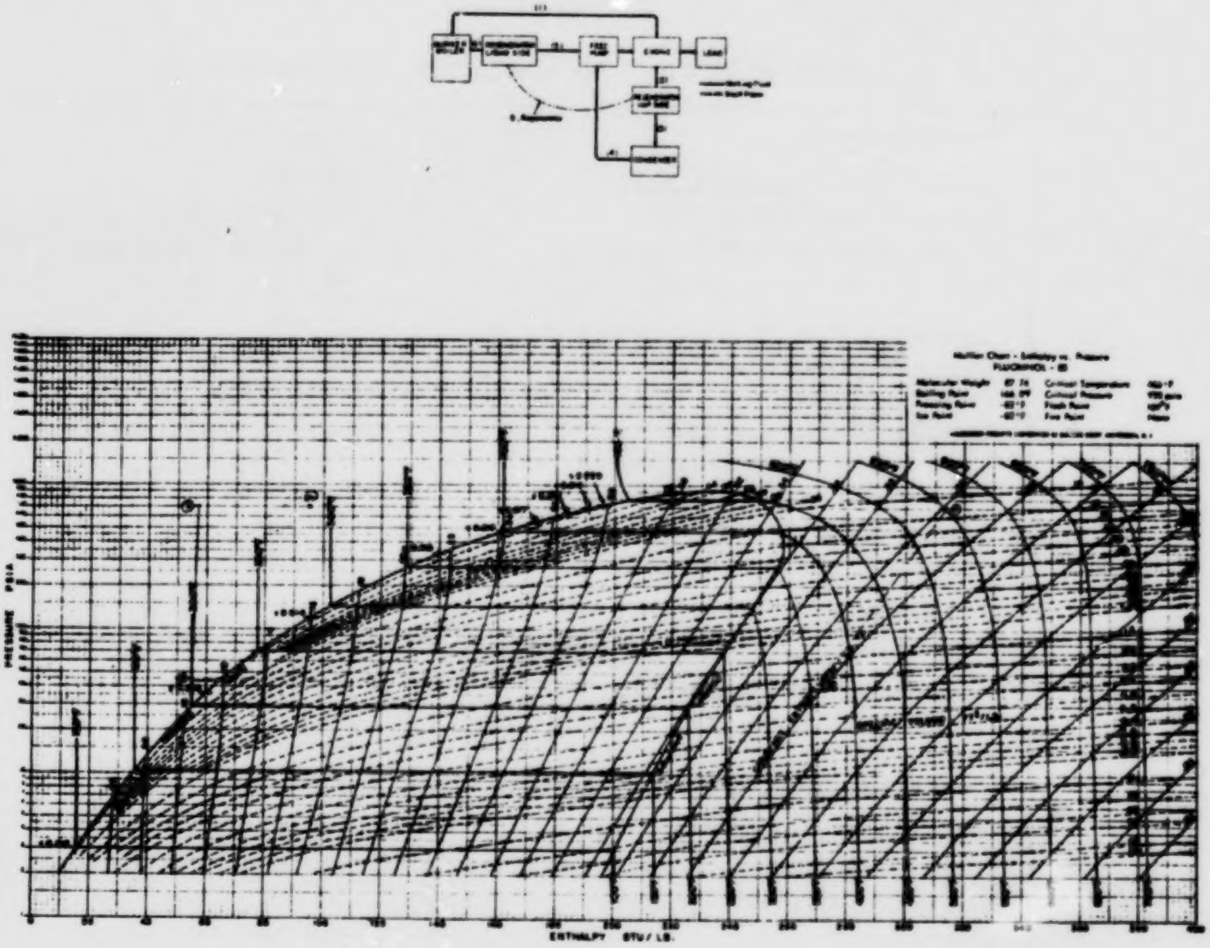


Figure 3. Pressure-Enthalpy Diagram and Cycle Conditions for Fluorinol-85.

TABLE 2

DESIGN POINT SPECIFICATIONS, FLUORINOL-85

Working Fluid	Fluorinol-85
Boiler Outlet Temperature	550°F
Boiler Outlet Pressure	700 psia
Boiler Heat Transfer Rate	1,600,000 Btu/hr
Boiler Efficiency (HHV)	82.5%
Expander Design Point Intake Ratio	0.137
Expander Displacement	107 in^3
Expander Speed	2400 rpm
Expander Piston Speed	1000 ft/min
Exp. Horsepower less Feedpump Power	103.2 hp
Expander Thermal Efficiency	84.6%
Expander Mechanical Efficiency	91.5%
Expander Overall Efficiency	77.5%
Regenerator Effectiveness	90%
Regenerator Heat Transfer Rate	276,000 Btu/hr
Condensing Temperature	217°F
Condensing Pressure	43 psia
Subcooled Liquid Temperature	197°F
Condenser Heat Transfer Rate	1,312,000 Btu/hr
Working Fluid Mass Flow Rate	7330
Organic Volumetric Flow Rate	11.70 gallons/minute
Feedpump Overall Efficiency	70.0%
Feedpump Power	6.4 hp
Cycle Efficiency	16.4%
Overall Efficiency	13.5%

TABLE 3

DESIGN POINT SPECIFICATIONS
THIOPHENE WORKING FLUID

Working Fluid	Thiophene
Boiler Outlet Temperature	550°F
Boiler Outlet Pressure	500 psia
Boiler Heat Transfer Rate	1.58×10^6 Btu/hr
Boiler Efficiency (HHV)	82.5%
Expander Design Point Intake Ratio	0.137
Expander Displacement	184 in^3
Expander Speed	2000 rpm
Expander Piston Speed	1000 ft/min
Expander HP less Feedpump Power	103.2 hp
Expander Thermal Efficiency	84.6%
Expander Mechanical Efficiency	91.5%
Expander Overall Efficiency	77.5%
Expander IMEP	127.4 psi
Regenerator Effectiveness	90.0%
Regenerator Heat Transfer Rate	0.249×10^6 Btu/hr
Condensing Temperature	216.2°F
Condensing Pressure	25.0 psia
Subcooled Liquid Temperature	96.2°F
Condenser Heat Transfer Rate	1.25×10^6 Btu/hr
Organic Mass Flow Rate	7377 pounds/hr
Organic Volumetric Flow Rate	[illegible] gallons/min
Feedpump Overall Efficiency	59.7%
Feedpump Power	5.25 hp
Cycle Efficiency	16.7%
Overall Efficiency	13.7%

since the details with Fluorinol-85 have not been completed. The differences are generally small; where differences occur with the Fluorinol-85, they are pointed out.

2.2 Expander Design

Establishing the dimensions of a new expander design depends, among other things, on the prediction of the indicated and mechanical efficiencies of the expander at the design condition. Consequently, a detailed analysis was carried out with thiophene to determine these efficiencies as functions of piston speed and load, using measured efficiencies obtained with the 5 hp expander on test at Thermo Electron as a check. The performance with Fluorinol-85 should be almost identical to the thiophene results. As a result of the analysis, it is possible to plot expander efficiency versus piston speed at various loads. One such plot is shown in Figure 4 for an indicated mean effective pressure (IMEP) of 125 psi. The rapid drop in efficiency with piston speeds above 1000 ft/min occurs due to inlet valve losses.

From this analysis, a piston speed of 1000 ft/min was selected for the design condition of 103 bhp at a vehicle speed of 95 mph. The reduction in expander size which could be realized by selecting a higher piston speed would probably be more than lost in boiler and condenser size increases due to lower overall cycle efficiency.

The IMEP and BMEP are determined by the cycle design condition, and the BMEP and the piston speed determine the piston area required to develop the desired horsepower. A 90°V of four cylinders was selected as being reasonably compact without either an excessive number of moving parts or excessive torque variation. The V design

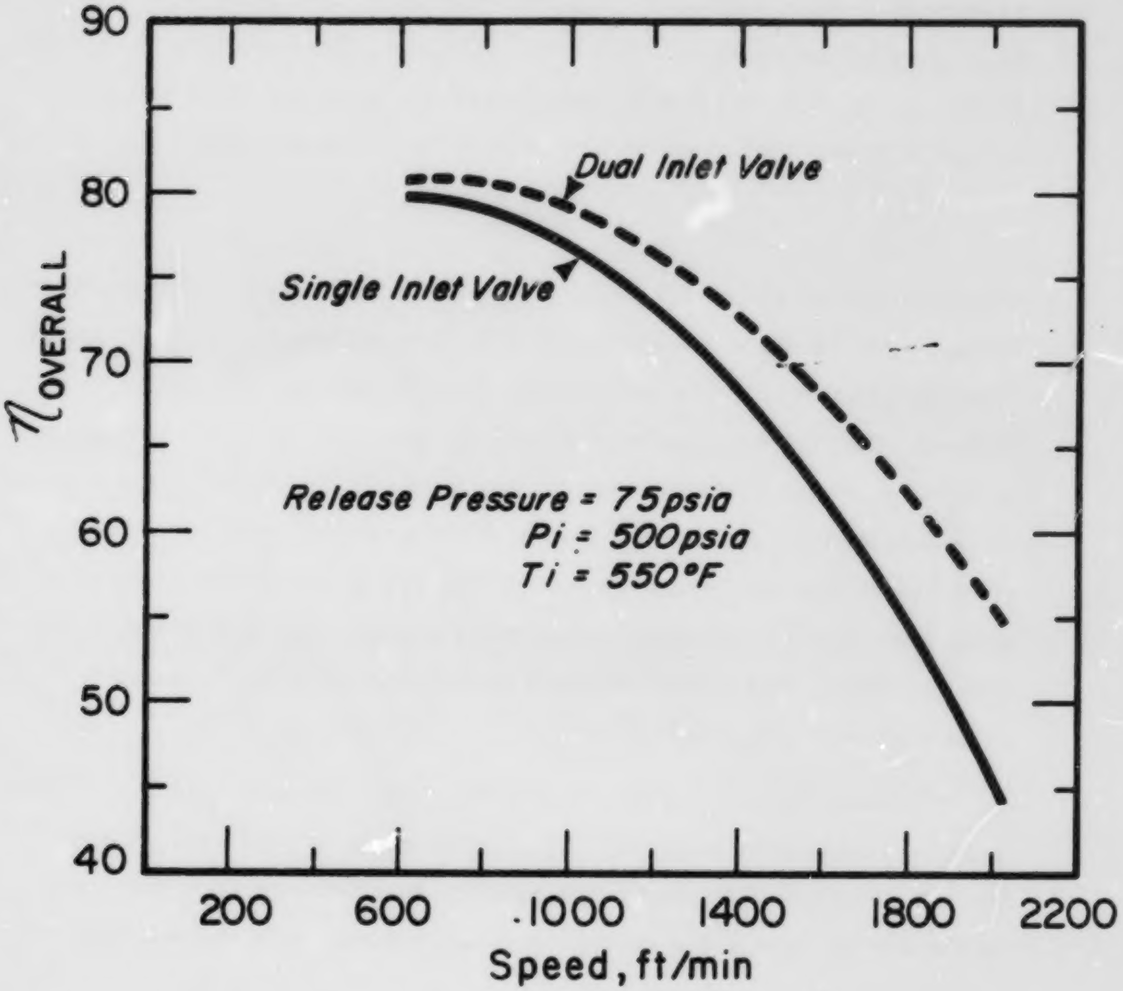


Figure 4. Overall Expander Efficiency Variation with Piston Speed

results in a short engine for a given number of cylinders and facilitates packaging of the system. With four cylinders, the resulting bore with Fluorinol-85 is 3.68 inches. The mean piston speed (1000 ft/min at design) and the engine speed are related by the expression

$$S = 2\ LN$$

where S = mean piston speed, L = stroke, and N = rpm. The selected design point speed of 2400 rpm, based on a reasonable bore-to-stroke ratio (1.47) and on valve train dynamics, results in a stroke of 2.5 inches. The basic expander dimensions and specifications are given in Table 4, with the values previously used for thiophene given for comparison; cross-sectional views of the expander with dimensions given for thiophene are presented in Figures 5 and 6 with hydraulic expander valving, slipping-clutch transmission, and feedpump incorporated. The overall expander dimensions will be reduced somewhat with Fluorinol-85.

All materials are identical to those now used in automotive internal combustion engines. The Rankine-cycle expander differs from the current automotive internal combustion engine in two very important aspects of its design: the inlet valving and the bearing design.

2.2.1 Variable Cut-Off Inlet Valving

The importance of having a valving system with variable cut-off is established in Section 3. Apart from the difficulty in varying the cut-off, the valving problem is considerably more severe than in internal combustion engines. To avoid excessive losses, the high

TABLE 4

EXPANDER DIMENSIONS AND SPECIFICATIONS

Working Fluid	Thiophene	Fluorinol-85
Configuration	Four Cylinders, 90°V	Four Cylinders, 90°V
Bore	4.42 inches	3.68 inches
Stroke	3.0 inches	2.50 inches
Displacement	184 in^3	107
BHP (feedpump work deducted)	103 at 2000 rpm	103 at 2400 rpm
IMEP at Design	127 psi	175 psi
BMEP at Design	117 psi	160 psi

TABLE 4

EXPANDER DIMENSIONS AND SPECIFICATIONS

Working Fluid	Thiophene	Fluorinol-85
Configuration	Four Cylinders, 90° V	Four Cylinders, 90° V
Bore	4.42 inches	3.68 inches
Stroke	3.0 inches	2.50 inches
Displacement	184 in^3	107
BHP (feedpump work deducted)	103 at 2000 rpm	103 at 2400 rpm
IMEP at Design	127 psi	175 psi
BMEP at Design	117 psi	160 psi

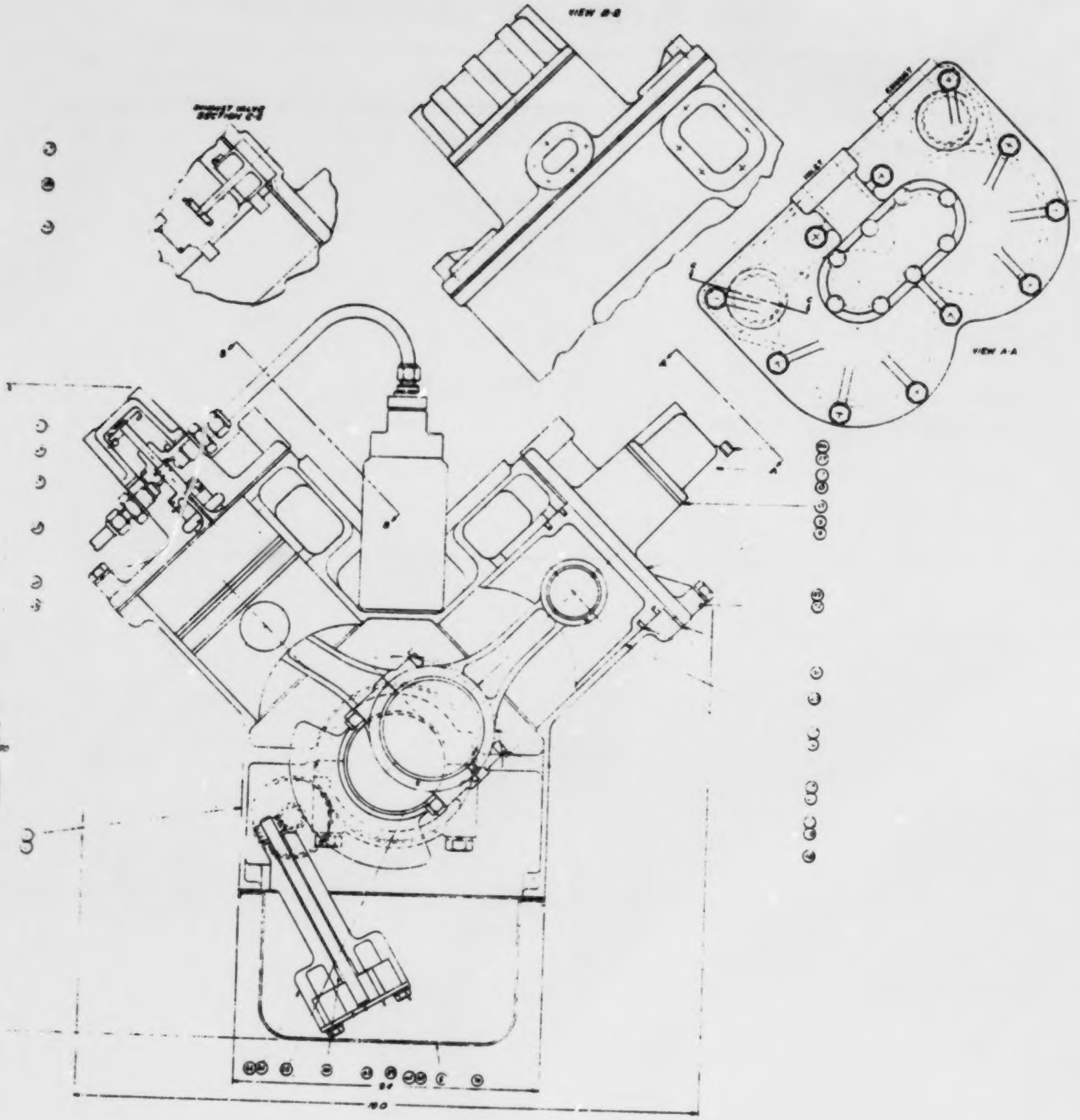


Figure 5. V-4 Expander with Hydraulically Actuated Valves, Front View.

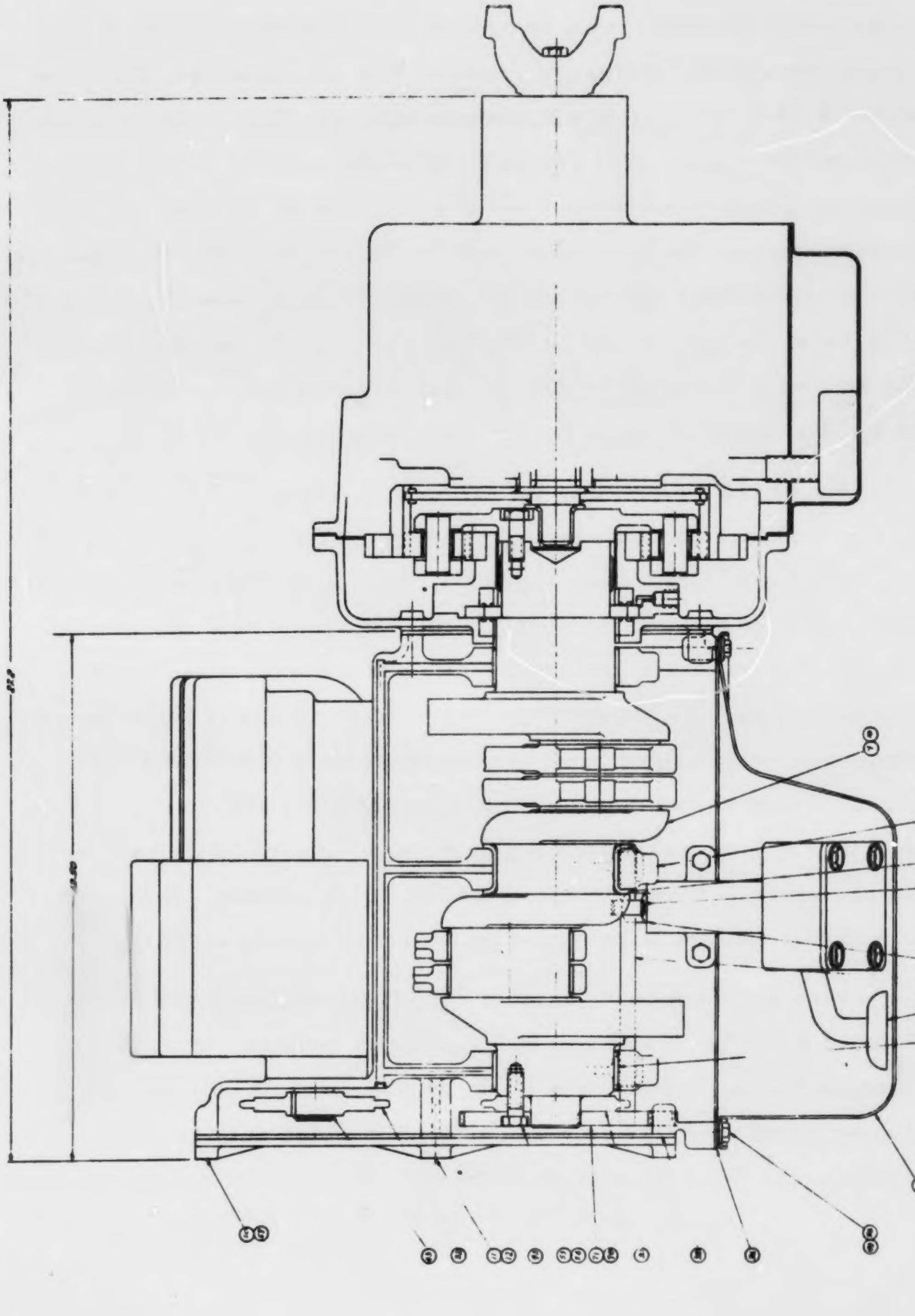


Figure 6. V-4 Expander with Hydraulically Actuated Valves, Side View.

density of the vapor at expander inlet conditions necessitates an inlet valve comparable in diameter and lift (and therefore mass) to the intake valve of an internal combustion engine. However, the valve event is much shorter in the Rankine expander than in the internal combustion engine. The design point intake ratio of 13.7% corresponds to a maximum valve event of 60°, whereas in internal combustion engines the inlet valve event is on the order of 240° or more. In a cam-operated system at a given speed, the acceleration and, therefore, the stress level are proportional to the lift divided by the square of the valve event; the cam stresses are much higher at a given expander speed for the vapor expander.

One way of overcoming the problem is shown in Figure 7. In this system two concentric inlet valves in series are driven by two separate camshafts. Cam number 1, driving inlet valve 1, has fixed timing with respect to the crankshaft. Cam number 2 has variable timing with respect to the crankshaft, and the total valve event is determined by the overlap of the two valves. In this way, relatively long cam events can be used, giving reasonable sized camshafts. Figure 7 also shows that the mean valve opening area can be higher with this approach than with a single valve, which should compensate for the lower flow coefficient of the two valve system. A modification of this concept is being set up for bench testing at TECO.

Other approaches to variable cut-off inlet valving are shown in Figures 8 and 9. These are both hydraulic devices. A directly-actuated hydraulic system is shown in Figure 8: A cam-operated plunger pump operates a hydraulic column which acts on a stepped piston on the inlet valve stem. The pump plunger is constructed with

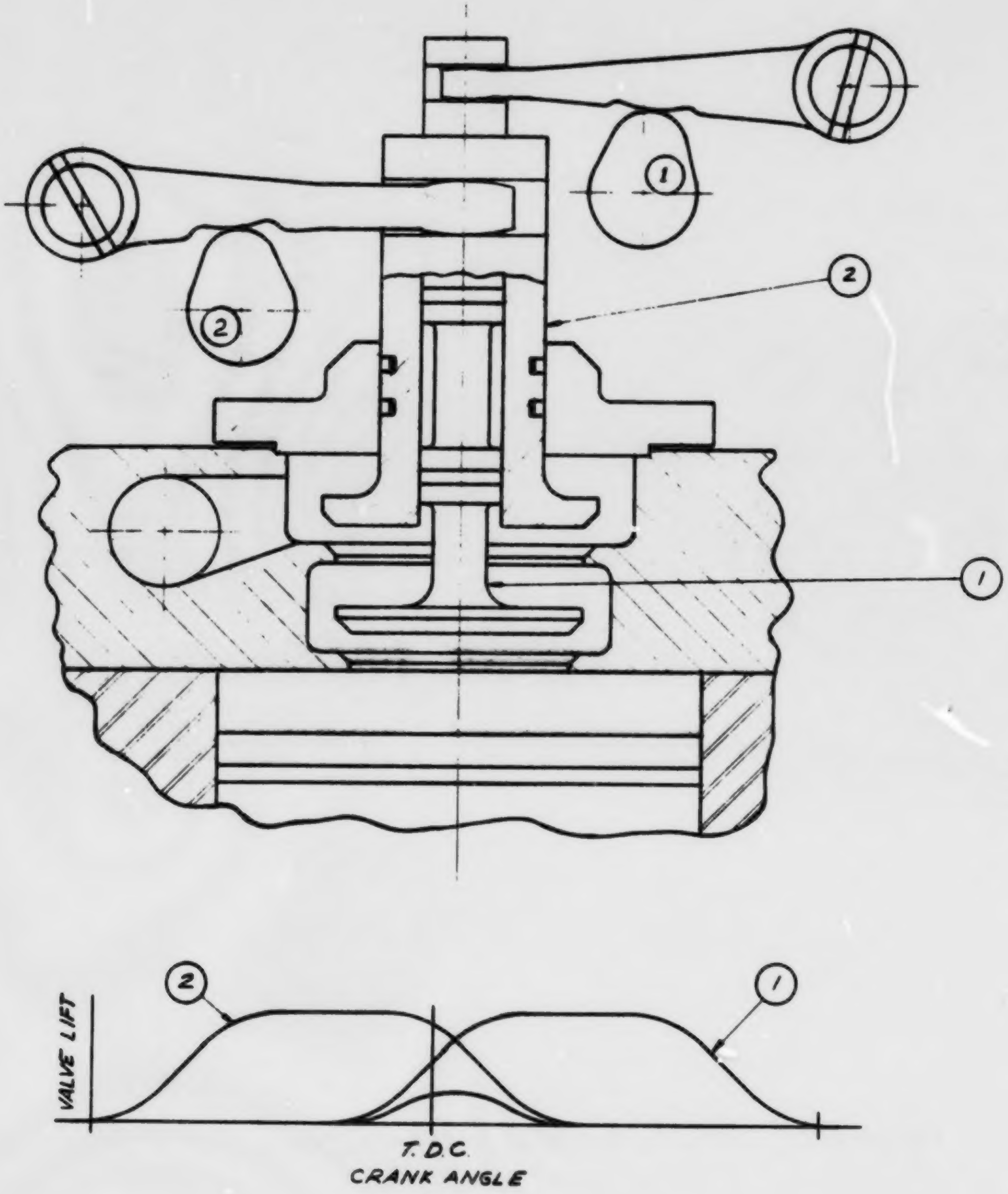


Figure 7. Two Inlet Valves in Series.

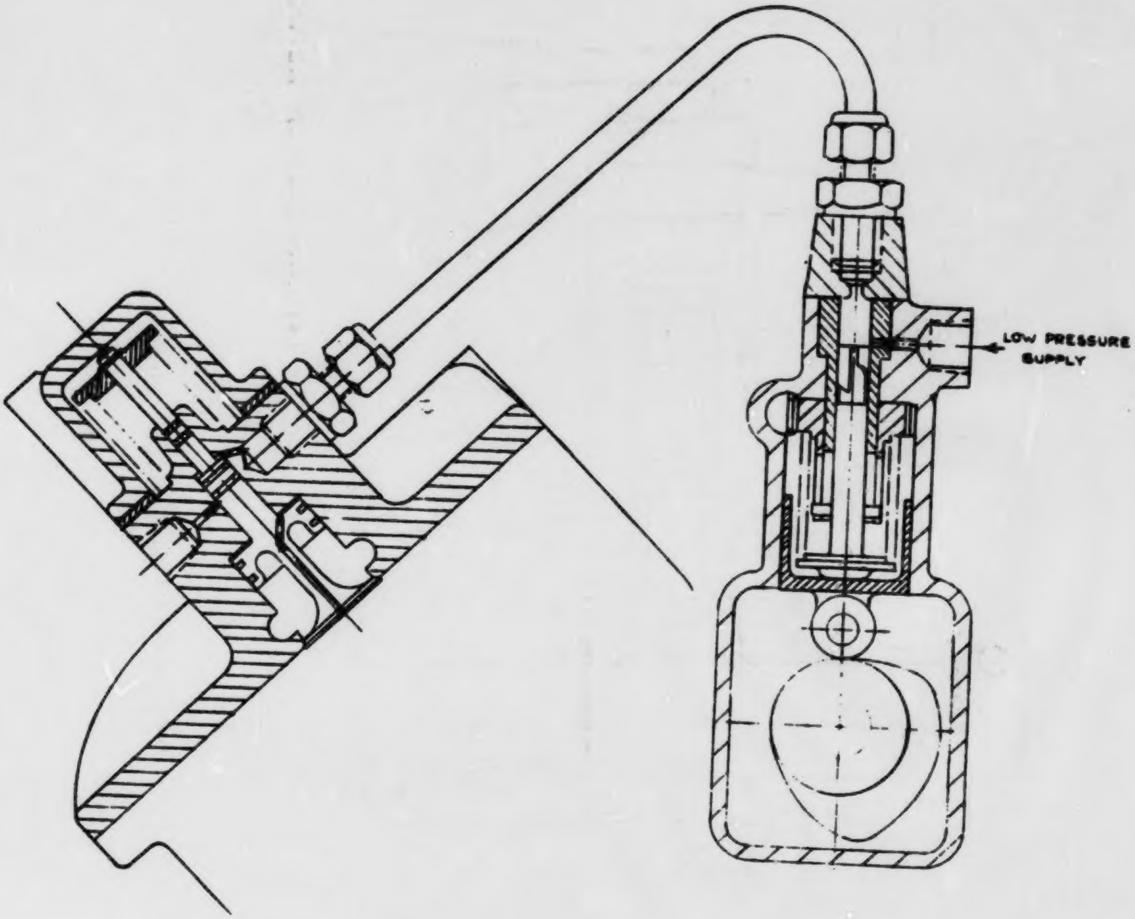


Figure 8. Directly Actuated Hydraulic Valve.

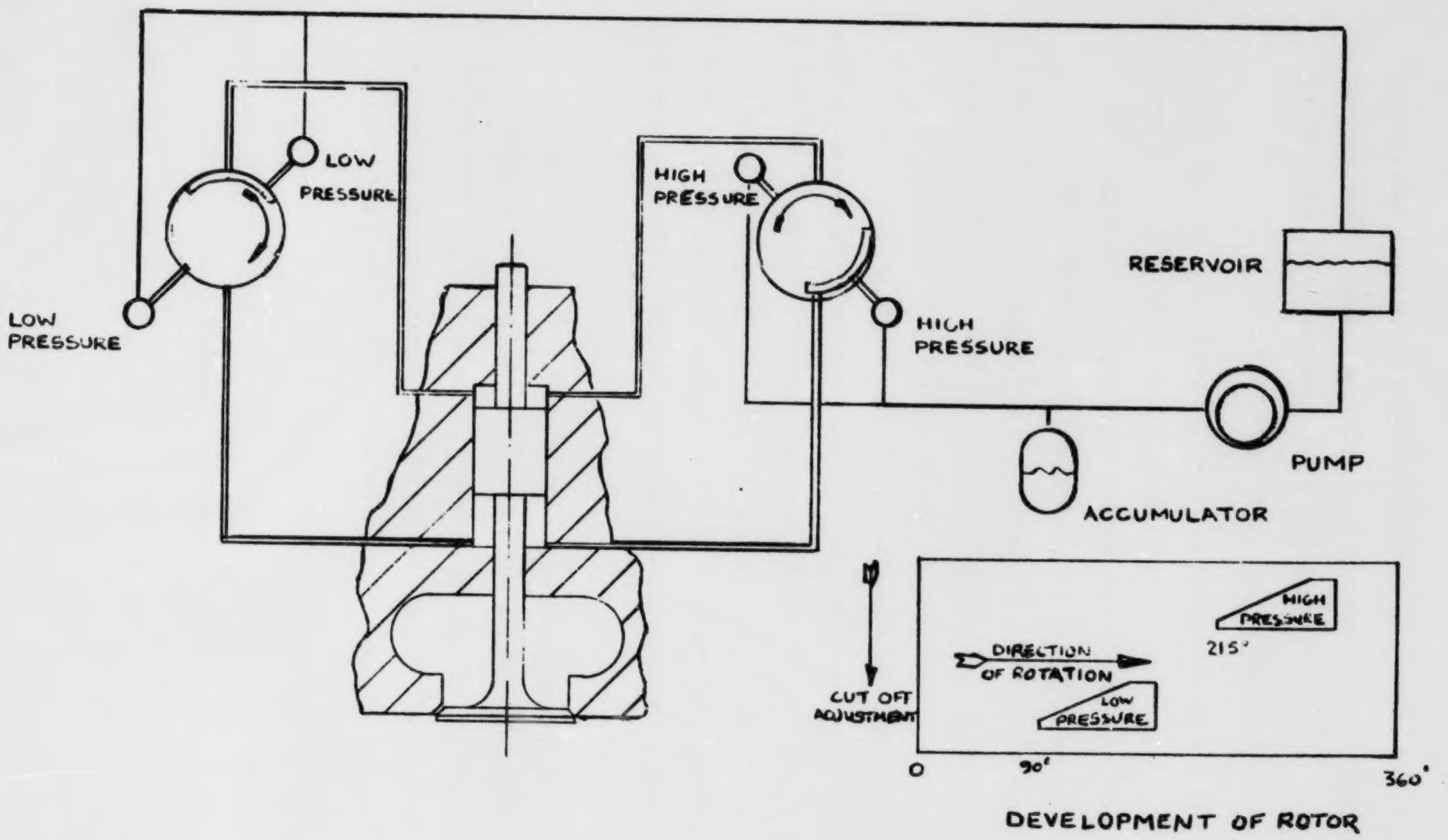


Figure 9. Pilot Operated Hydraulic Valve.

a helical undercut so that its angular position in its bore determines its effective stroke, thus varying inlet valve duration. This system is quite similar to diesel engine injection systems but has the disadvantage of a relatively large power requirement.

Another hydraulic scheme is shown schematically in Figure 9. In this system, a pump supplies high pressure oil at 700 - 1000 psi to a rotary valve (shown as two valves for simplicity in Figure 9). The rotary valve supplies the high pressure oil to alternate sides of a piston connected to the inlet valve. Cut-off adjustment is obtained by moving the rotary valve axially in its housing. Thermo Electron Corporation has contracted with the British Internal Combustion Engine Research Institute for fabrication and bench testing of a full-size valving system based on this concept. A subcontract to American Bosch Corporation has also been arranged under Phase II of the APCO program at TECO for bench testing of a similar concept which uses a solenoid-operated pilot valve in place of a rotary valve. Both of these approaches have a low power requirement of about one horsepower for a 4 cylinder expander.

It is thus expected that bench test results from three different approaches will be available for selection of the expander intake valving system to be used in the system. Phase II of the APCO program also involves bench-testing of a full-size exhaust valve scaled up from the exhaust valve used on the TECO 5 hp expander.

2.2.2 Bearing Design and Selection

The expander bearing design is strongly influenced by the type of transmission used. If the expander is coupled directly to the driveshaft, as in many early steam cars, journal bearings relying on hydrodynamic lubrication cannot be used; roller or ball bearings must be used, because of the high bearing loads which could occur at essentially zero rpm. On the other hand, if a conventional torque converter were used, bearing sizes, at least on the crankshaft, can be comparable to an internal combustion engine of the same bore, since the peak cylinder pressures are roughly the same and the inertia loading is lower on the Rankine expander (because of the 1000 ft/min limit on piston speed). The wrist pin bearing is more heavily loaded than in the conventional four-stroke internal combustion engine because the load on the pin never reverses; in this respect, it is much like the wrist pin in a two-stroke engine. Any single speed transmission can load the expander bearings fairly heavily when the expander is idling at 360 rpm and the clutch is engaged. At these conditions, with a maximum intake ratio of 80%, the bearing loading is such that bearing sizes associated with a two-stroke diesel of the same bore would be barely adequate for the Rankine expander. A transmission with two forward speeds as well as a reduction in the maximum intake ratio, as planned for the system, will alleviate this situation.

It is expected that, with development, conventional journal bearings can be used in the expander if the transmission used permits the expander to idle at zero vehicle speed. However, to minimize the development problems and the consequences of momentary lubrication failure on the system, needle bearings will be used throughout the expander for the first prototypes. Needle bearings have been used in the 5 hp expanders tested at Thermo Electron Corporation with excellent reliability.

2.3 Feedpump Design

A full-size feedpump sized for the thiophene is under loop-test at TECO as part of the Phase II of the APCO program. This pump was designed and constructed at TECO. The primary factors considered in the selection and design of the vapor-generator feedpump are that the pump must be positive displacement because of the high discharge pressure. The lubricity of either Fluorinol-85 or thiophene is relatively poor and its liquid viscosity low. The pumping rate must be variable from basically zero to 15 gpm for thiophene over a 800 - 2000 rpm range, and the pump must operate with low net positive suction head (NPSH) without cavitation, since the NPSH is provided primarily by subcooling of the liquid coming from the condenser.

The feedpump selected, illustrated in Figure 10, is a 5-cylinder piston pump driven by a wobble plate; its characteristics are summarized in Table 5. The selection of a piston pump was based on testing of several types of positive displacement pumps, including gear and vane-type pumps, at Thermo Electron. In general, high leakage rates (low volumetric efficiency) and high wear rates have been encountered for all pump types other than piston, which have given completely satisfactory performance. With the piston pump all bearing surfaces can be oil-lubricated.

The variable pump rate is obtained by incorporation of variable displacement in the feedpump, permitting the pumping rate to be controlled at the desired rate regardless of feedpump speed. The method used to obtain variable displacement is to vary the effective displacement of the feedpump by bypassing all or part of the pump output to the condenser. The effective displacement is obtained by moving the

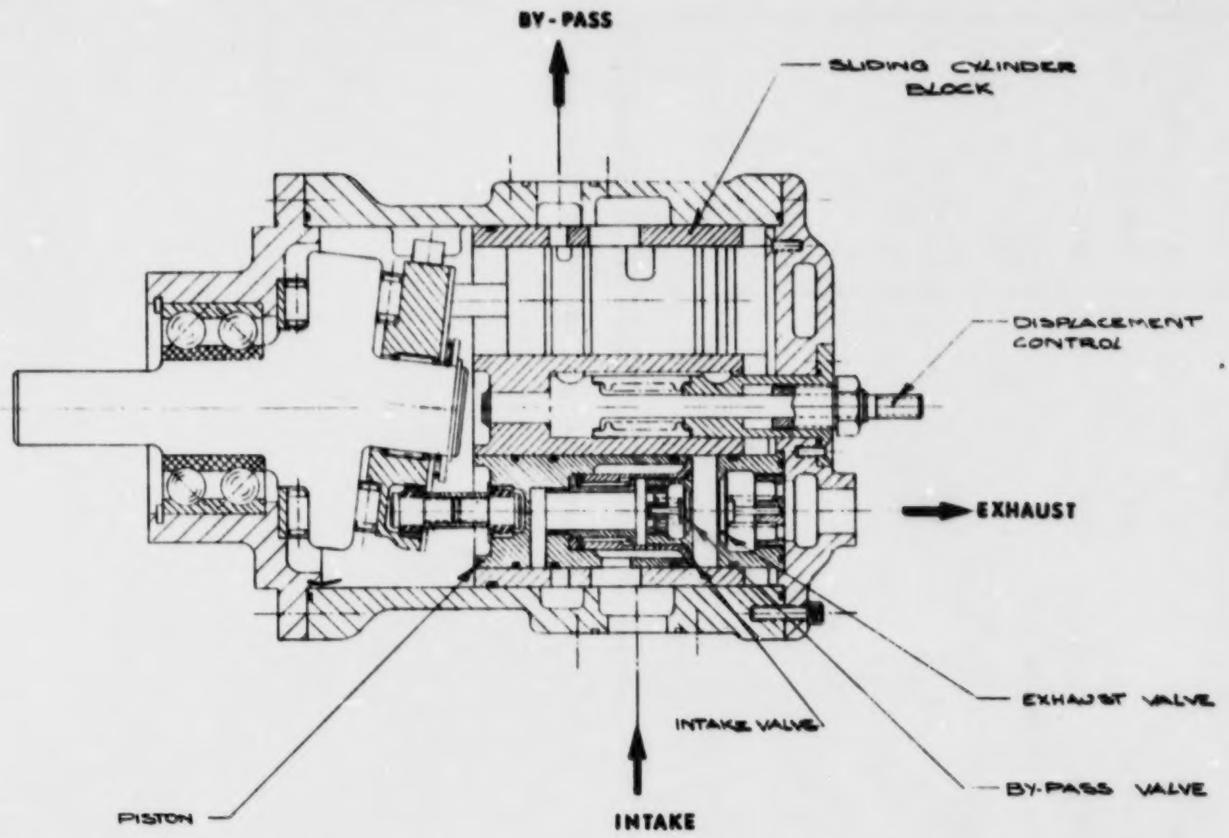


Figure 10. Feedpump Cross Section.

TABLE 5

FEEDPUMP CHARACTERISTICS

Working Fluid	Thiophene	Fluorinol-85
Pumping Rate	15 gpm	11.7 gpm
Number of Cylinders	5	5
Volumetric Efficiency	90%	90%
Overall Efficiency	80%	80%
Range for Maximum Pumping Rate	800-2000 rpm	960-2400 rpm
Total Displacement	5.51 in^3	3.58 in^3
Bore	1.875 in	1.51 in
Stroke	0.4 in	0.4 in
Materials of Construction		
Housing	Cast Iron	Cast Iron
Pistons and Valves	Hardened Steel	Hardened Steel
Bearings	Needle	Needle

entire cylinder block to regulate the axial position at which the bypass port is uncovered during the discharge stroke of each of the five pistons The bypass porting is configured so that pumping to the high pressure exhaust occurs always from bottom dead center; this procedure minimizes the discharge pressure transients and noise level of the pump. The bypass flow is also never pumped to the high pressure, thereby minimizing the feedpump parasitic load on the system. The control lever for positioning the cylinder block uses a bellows rather than a sliding seal to maintain the hermetic nature of the system.

Spring-loaded poppet suction and discharge valves are used. The suction valve is constructed in the cylinder and is made as large as possible to minimize the pressure loss through the valve and the tendency for cavitation. The smaller discharge valve is located in the cylinder head. Common suction and discharge plenums for all five cylinders are incorporated in the housing castings. The use of five cylinders was based on reducing pressure transients due to the flow variation from the piston pump. These transients must be maintained sufficiently small on the suction side of the pump that the liquid pressure never falls below the vapor pressure of the subcooled liquid. A computer analysis of the pressure transient behavior indicated that a five-cylinder pump would be required to prevent cavitation with 20°F subcooling at the pump suction.

The pump drive could be either crank or wobble-plate. The wobble-plate drive was selected because of its compactness and easier packaging with the expander, its lower weight and vibration, its quieter operation at higher speeds, and its more convenient geometry for variable displacement incorporation.

2.4 Burner-Boiler Design

The burner-boiler design has not been worked out in detail for Fluorinol-85. In Figures 11, 12 and 13 the boiler tube bundle design and the burner cross sections are presented, based on detailed computer analysis for the thiophene fluid. The design requirements are summarized in Table 6. Some modification will be necessary for Fluorinol-85, due to the larger superheat requirement for Fluorinol-85 relative to thiophene. The changes will be restricted to modifications of the tube bundle in each stage, with no change in the overall burner-boiler envelope or in the flow paths in the boiler. The factors considered in arriving at the boiler design, in addition to heat transfer performance, were low materials cost, low volume, easy construction, and low combustion side pressure drop. Maintaining a low combustion side pressure drop is of extreme importance in meeting the development goals of fast startup and low parasitic power loss. For startup, the complete combustion system must be electrically driven by the battery. To achieve fast startup, it is essential that the burner be operated at its maximum rate. The total peak power requirement must therefore be maintained at as low a level as possible, consistent with the packaging envelope and heat transfer rating required. The current design at full burning rate requires about 1-1/4 hp total to operate the combustion air blower and fuel nozzle compressor. This power level is practical considering the battery, alternator and motor sizes required.

With reference to Figure 11, the combustion gases, at a temperature of ~ 3300°F, flow from the combustion chamber into the center of the tube bundle and radially outward through the tube bundle. The flow

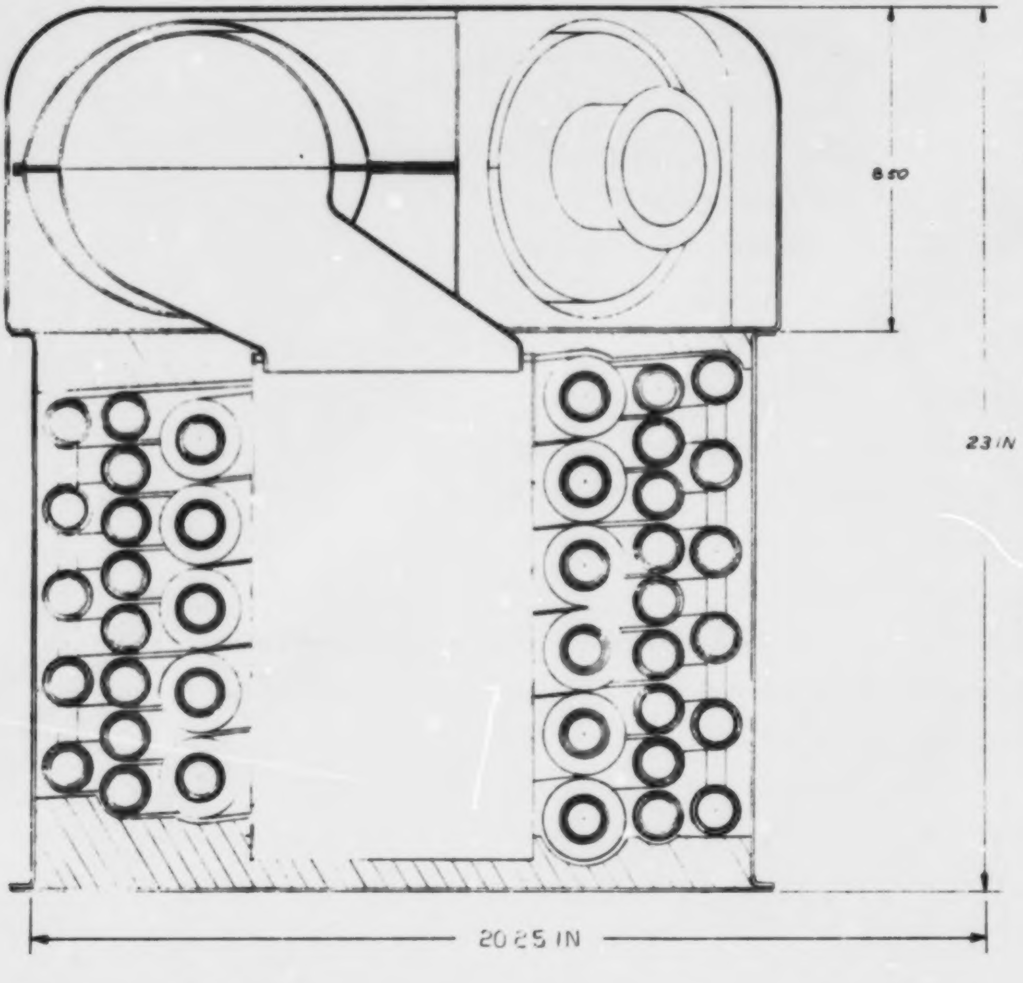


Figure 11. Cross Section Through Burner-Boiler, Short Axis.

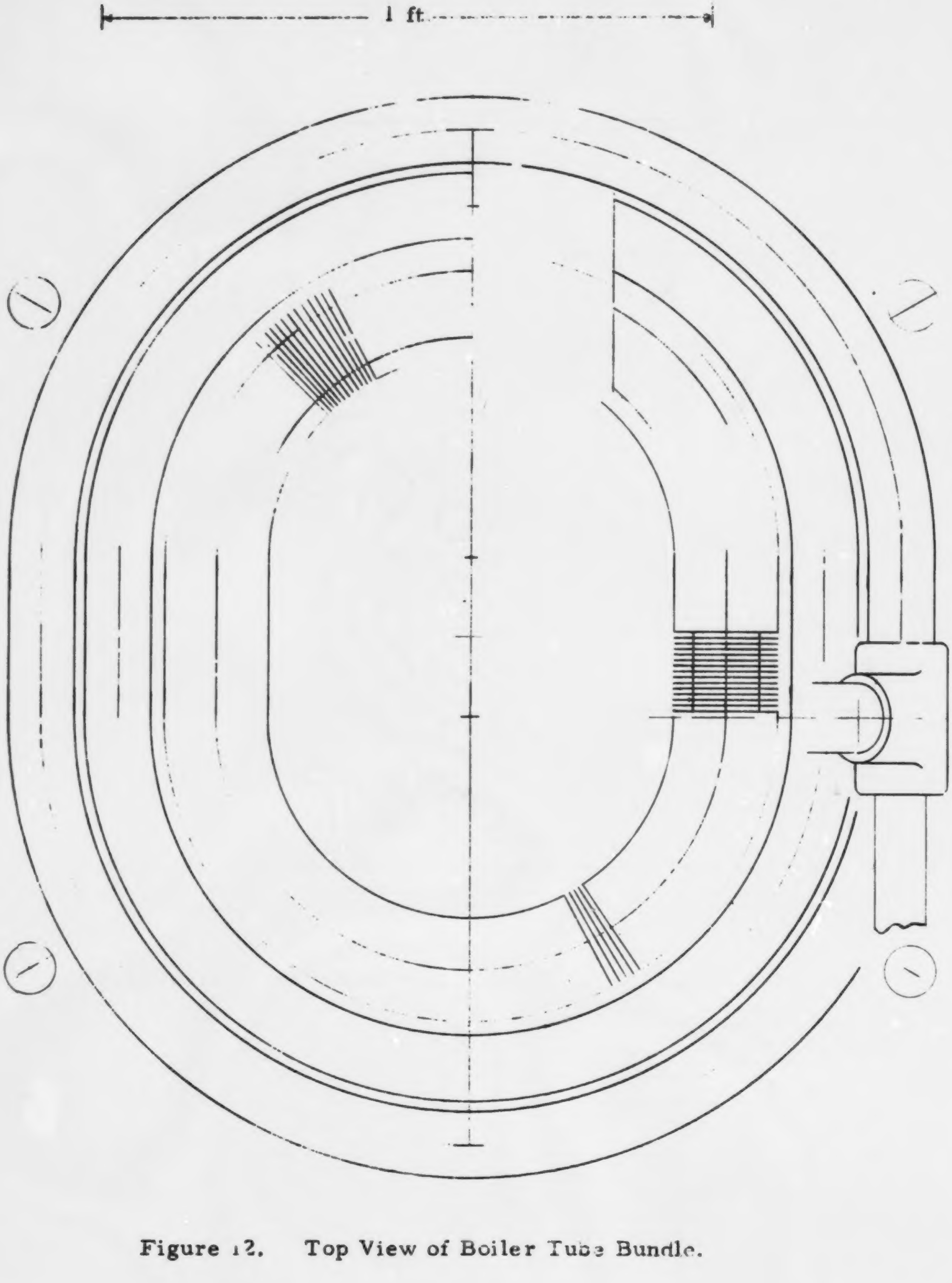


Figure 12. Top View of Boiler Tube Bundle.

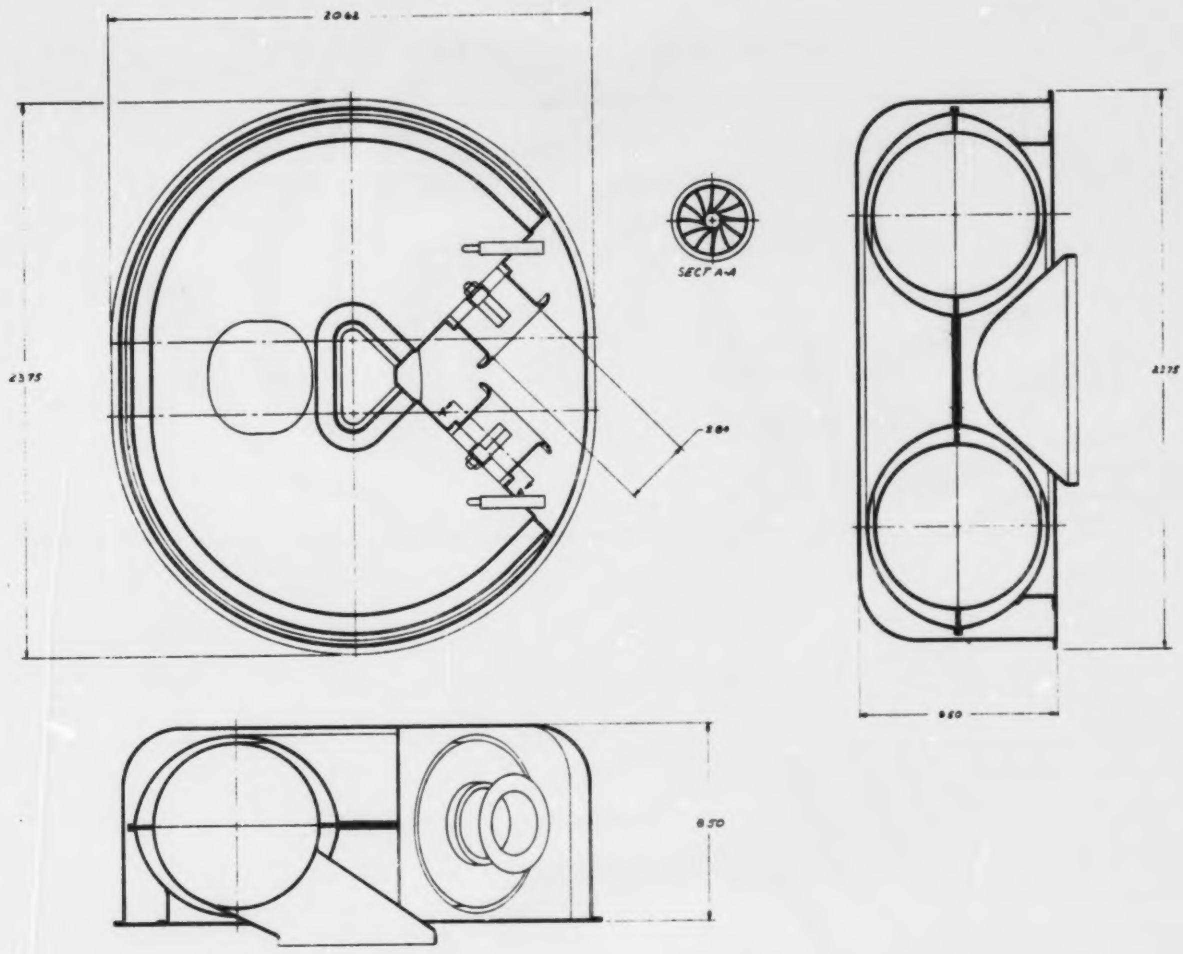


Figure 13. Cross Sections Through Automotive-Size Burner.

TABLE 6

BURNER-BOILER DESIGN REQUIREMENTS

Reference Cycle Boiler Heat Transfer Rate	1.60 x 10^6 Btu/hr
Maximum Boiler Heat Transfer Rate	1.70 x 10^6 Btu/hr
Burner Design Maximum Heat Release Rate (HHV)	2.06 x 10^6 Btu/hr
Boiler Design Efficiency (HHV)	82.5%
Turndown Ratio	15/1

path of the organic through the tube bundle is illustrated in Figure 14. The organic first flows through stage 3, from which the combustion gases are exhausted; this provides the lowest organic temperatures in the boiler at the combustion gas outlet and a high boiler efficiency without air preheat. It is important that an extremely compact and efficient heat transfer surface be used in this stage to maximize the boiler efficiency with acceptable pressure drop on the combustion side. The organic next flows through the inner stage (through which the combustion gases first flow), with a resultant high heat transfer coefficient. Because of the high gas temperature and extended surface on the combustion side, coupled with the high heat transfer coefficient on the organic side, a very high heat transfer rate can be obtained in the first stage. The organic next flows through the superheater coil or stage 2. This stage is a bare tube coil, since the controlling thermal resistance is on the organic side and an extended heat transfer surface is not required on the tube.

The characteristics of the three boiler stages are given in Table 7; Figure 15 presents the calculated design point temperature and pressure profiles through the boiler. In the last or third stage, a matrix made of steel and copper balls brazed together and to the tube is used. This type of extended surface provides a very high heat transfer rate per unit volume and is amenable to mass production techniques. Thermo Electron Corporation is currently making heat transfer and pressure drop measurements on a full-size section of the boiler third stage as part of Phase II of the APCO program. Other types of extended surface for the boiler third stage will also be evaluated experimentally.

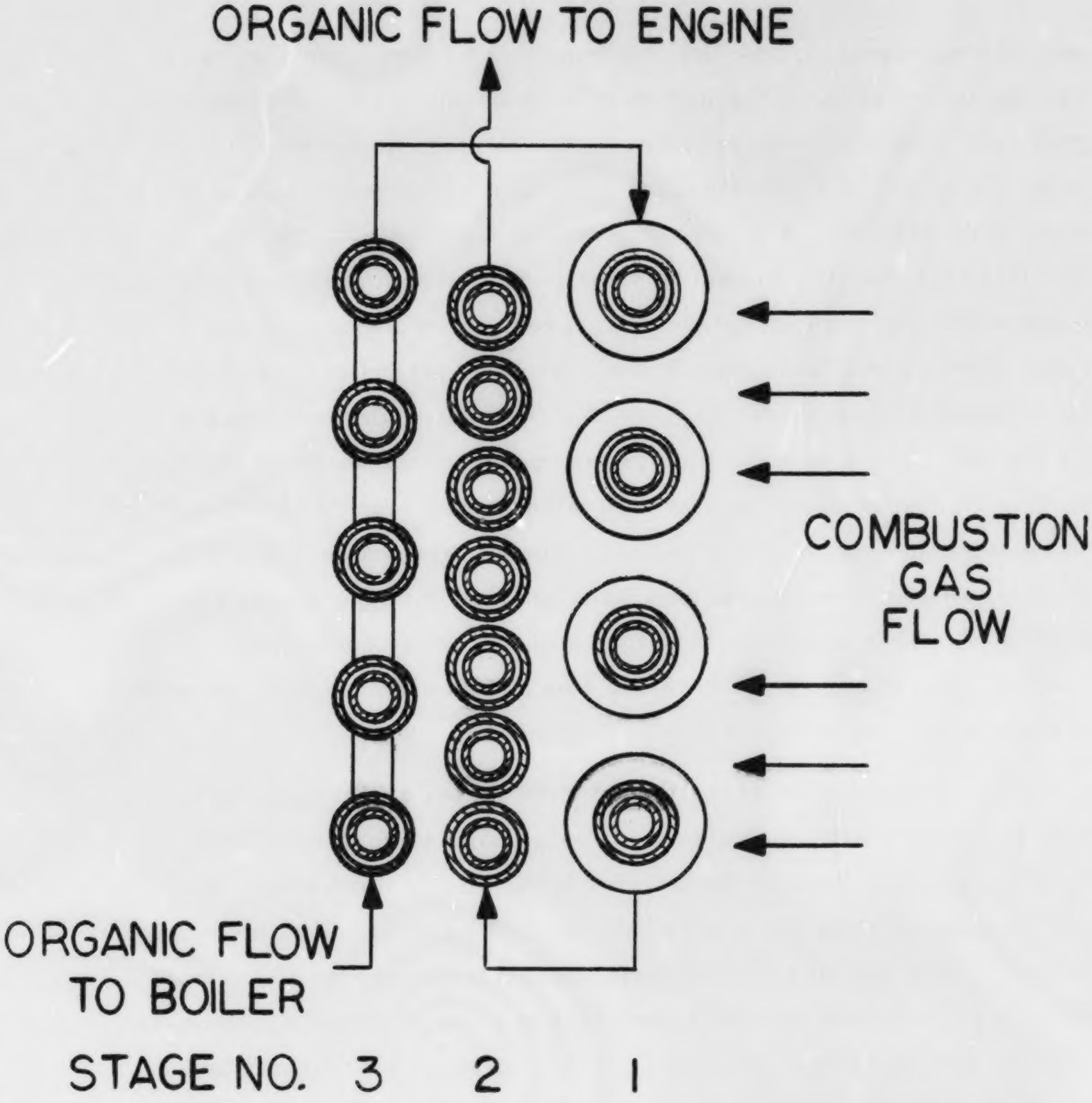


Figure 14. Organic Flow Path through Boiler Tube Bundle

TABLE 7

BOILER DESIGN SPECIFICATIONS

Stage No.	Heat Transfer Rate Btu/hr	Combustion Gas Temp. °F		Tubing Length ft.	Pressure Drop	
		entering	leaving		Combustion Side in w. c.	Organic Side psi
1	1.05×10^6	3330	1670	17	0.17	—
2	0.33×10^6	1670	1114	35	0.33	—
3	0.36×10^6	1114	467	27	2.26	—
Total	1.74×10^6	—	—	79	2.76	46

Tube Specifications

Inner Tube ID = 0.930"
OD = 1.000"
Outer Tube ID = 1.125"
OD = 1.315"

Fin Specifications

Fins/Inch	10
Fin Thickness	0.012"
Fin Material	Copper
Fin Height	0.356"

Matrix Specifications

Ball Size	3/32"
Ball Material	50% Carbon Steel, 50% Copper
Matrix Thickness	0.5"
Matrix Height (between tubes)	0.970"

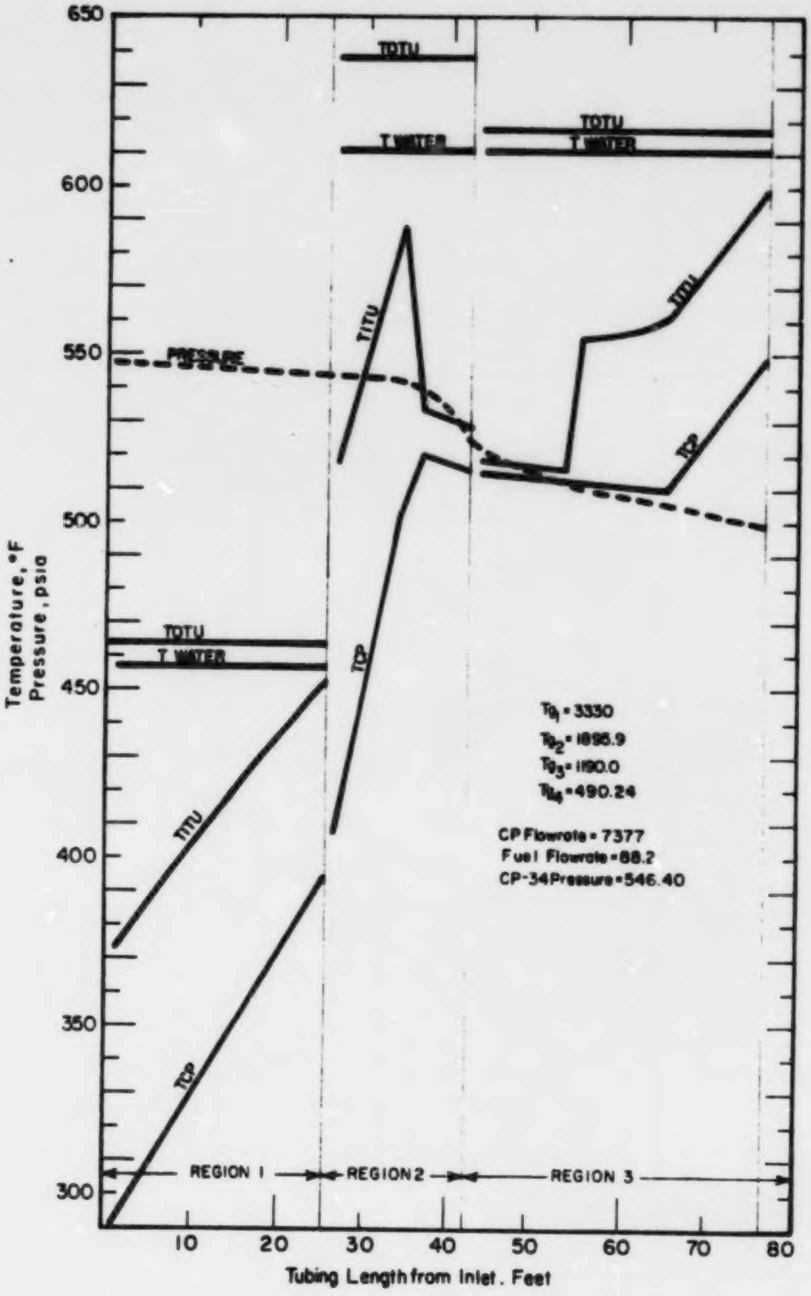


Figure 15. Design Point Boiler Temperature Profile

An intermediate heat transfer fluid (water) is used, with double tube construction in the boiler tubes, to positively prohibit hot spots on the organic side of the boiler. Stages 1 and 2 are connected to the same water reservoir, which represents the high pressure side of the boiler. The water side of stage 3 is separate and represents the low pressure water side of the boiler.

The combustion chamber design, illustrated in Figure 13, is based on a volumetric burning rate of 2.8×10^6 Btu/hr/ft^3 and is scaled from the burner used in the 5 hp system currently under test. The burner is constructed integral with the boiler tube bundle, as illustrated in Figure 11. To reduce the pressure drop within the diametrical clearance available, two identical burners operating in parallel are used rather than one longer burner with the same combustion chamber diameter. The pressure drop at maximum firing rate for the two-burner setup is 1.5" w.c.

While no pollution measurements are available on the full-scale burner design illustrated, Thermo Electron Corporation has completed measurements on a smaller scale burner (140,000 Btu/hr) with performance characteristics similar to the burner illustrated in the design. This burner is being used on a 5 hp system now on test at Thermo Electron Corporation. Figures 16 and 17 present the steady state emission levels from this burner as a function of excess air for burning rates of 105,000 Btu/hr and 50,000 Btu/hr, respectively. It is apparent that the emission levels are extremely low. To indicate the transient performance of the burner, the burner was oscillated between 50,000 and 105,000 Btu/hr burning rates with constant fuel-to-air ratios maintained; CO and unburned hydrocarbon emission levels

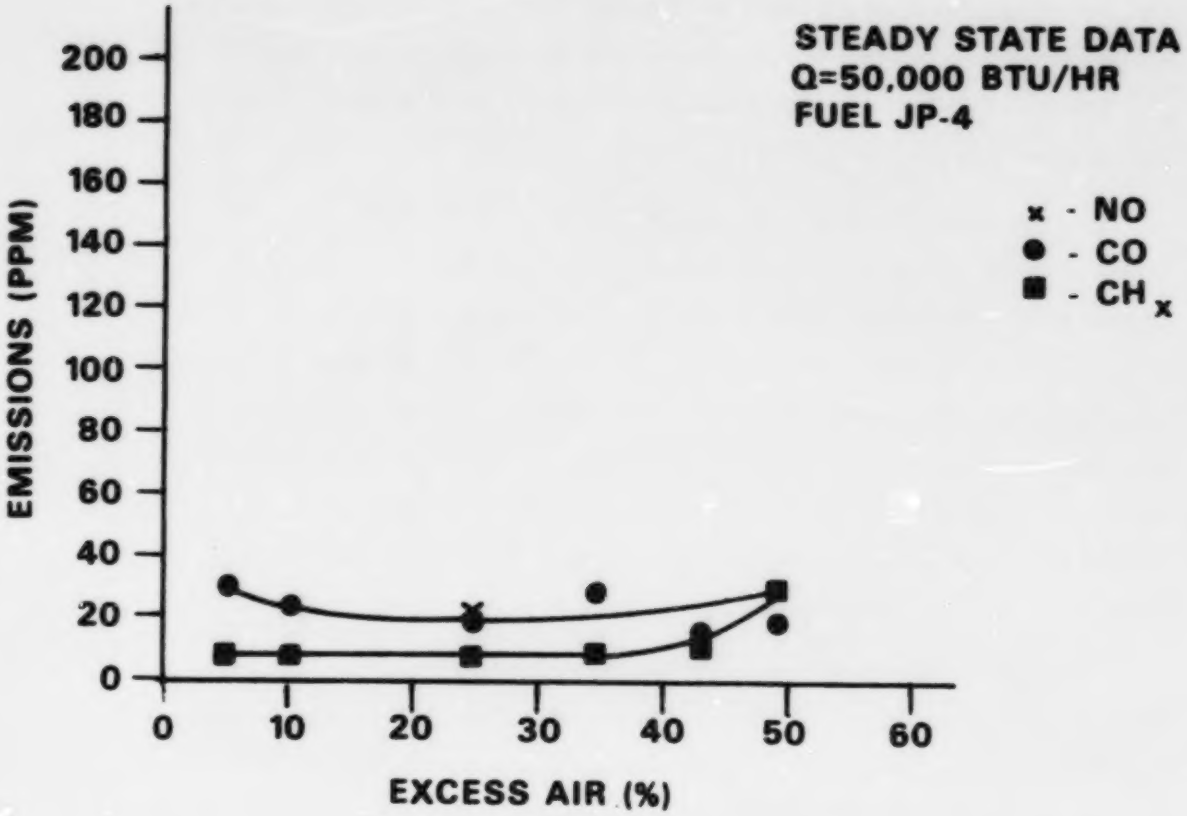


Figure 16. Effect of Excess Air on Emissions, 50,000 Btu/hr

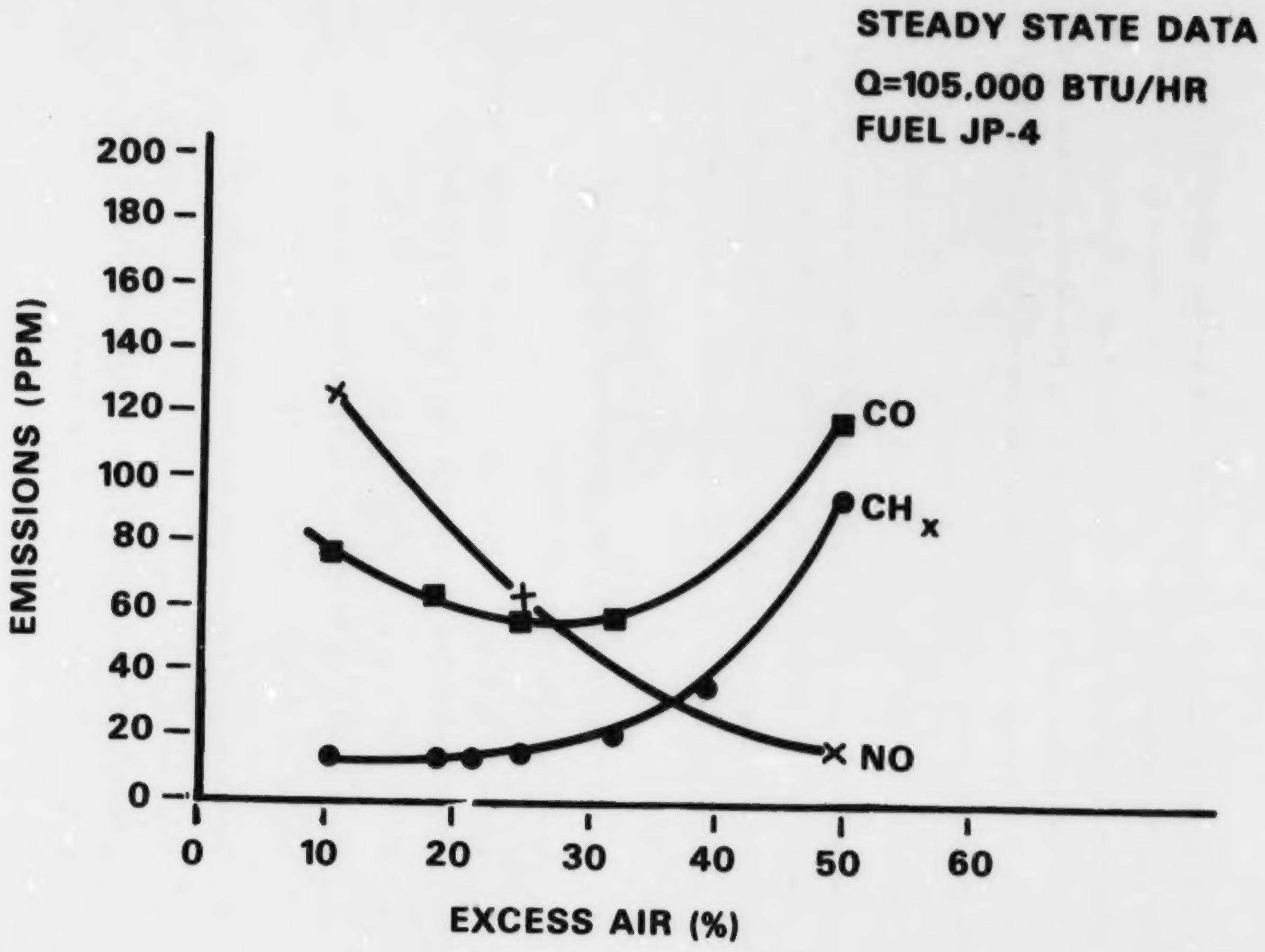


Figure 17. Effect of Excess Air on Emissions, 105,000 Btu/hr

were monitored continuously while a bag sample was collected throughout the run for NO measurement at the end. The results are indicated in Table 8. As indicated in Section 3, use of these emission concentrations with the system performance gives gm/mile emission levels significantly less than projected limits under the recently enacted Clean Air Legislation in the U. S.

2.5 Condenser Design

The condenser fan power represents the largest parasitic load of the system. A very important goal in the design of the component is to minimize the required fan power. The factors considered in the condenser design, as well as its integration and packaging with the system, were:

a. An efficient (low f/j ratio) extended surface should be used on the air side of the exchanger.

b. The organic side should be circuited to provide high vapor velocities leading to high organic side heat transfer coefficients.

c. The frontal area of the condenser (as well as the air free flow area) should be made as large as possible.

d. Highly efficient condenser fans should be used.

e. The system packaging and vehicle modifications should be designed to minimize grille (intake) and engine compartment (exhaust) air pressure losses.

TABLE 8

TRANSIENT EMISSION DATA

FIRING RATE (BTU/HR)	EXCESS AIR (%)	CH_x (PPM)	CO (PPM)	NO_x (PPM)	ELAPSED TIME (MIN)
105,000	25%	6	60	-	0
50,000	25%	-	-	-	.5
50,000	25%	5	30	-	1.0
50,000	25%	-	25	-	1.5
50,000	25%	4.5	70	-	2.0
105,000	25%	40[1]	180[1]	-	2.5
105,000	25%	7	90	-	3.0
105,000	25%	5	-	-	3.5
50,000	25%	5	80	-	4.0
50,000	25%	4	25	-	4.5
50,000	25%	4.5	20	-	5.0
50,000	25%	-	-	-	5.5
105,000	25%	15[1]	1000+[1]	-	6.0
105,000	25%	-	-	-	6.5
105,000	25%	6	75	-	7.0
50,000	25%	-	75	-	7.5
50,000	25%	4.5	30	-	8.0
50,000	25%	4	15	-	9.0
				42[2]	
				39[2]	

NOTES:

1. Short duration peak ∼ 10 sec.
2. Exhaust gas sample collected during the 9 minute run. Then two samples were drawn and NO analysis performed.

f. The condenser fan control should be optimized to minimize the fan power requirement by reducing the fan power when not required. The condenser fans are directly driven by the propulsion engine.

The condenser core of the conceptual design is illustrated in Figure 18 with the design point characteristics, as obtained with the condenser computer program, given in Table 9.

The condenser core is similar to a Ford radiator with louvered fins, except that the flattened tubes have a heavier wall (0.030" vs 0.005") and are constructed in one integral piece with partitions used to provide the desired vapor-side flow path. Copper fins with 2.5 mil thickness are used, and the tubing is made of carbon steel rather than brass as in the Ford radiator. The frontal area used in the design represents the maximum practical area for an intermediate size American car with some rework of the front end frame and grille.

It is expected that improvements can be made to the condenser design illustrated here. Use of a strip fin rather than the louvered fin results in an improved f/j ratio and will reduce the fan power required for a given condenser physical size. If the fluid compatibility properties are satisfactory, use of an all-aluminum condenser will significantly reduce the condenser weight and will permit easier construction.

2.6 Regenerator Design

The regenerator design conditions are illustrated in Figure 19 and the regenerator design in Figure 20. While the design is based on thiophene, the requirements are very similar to those for

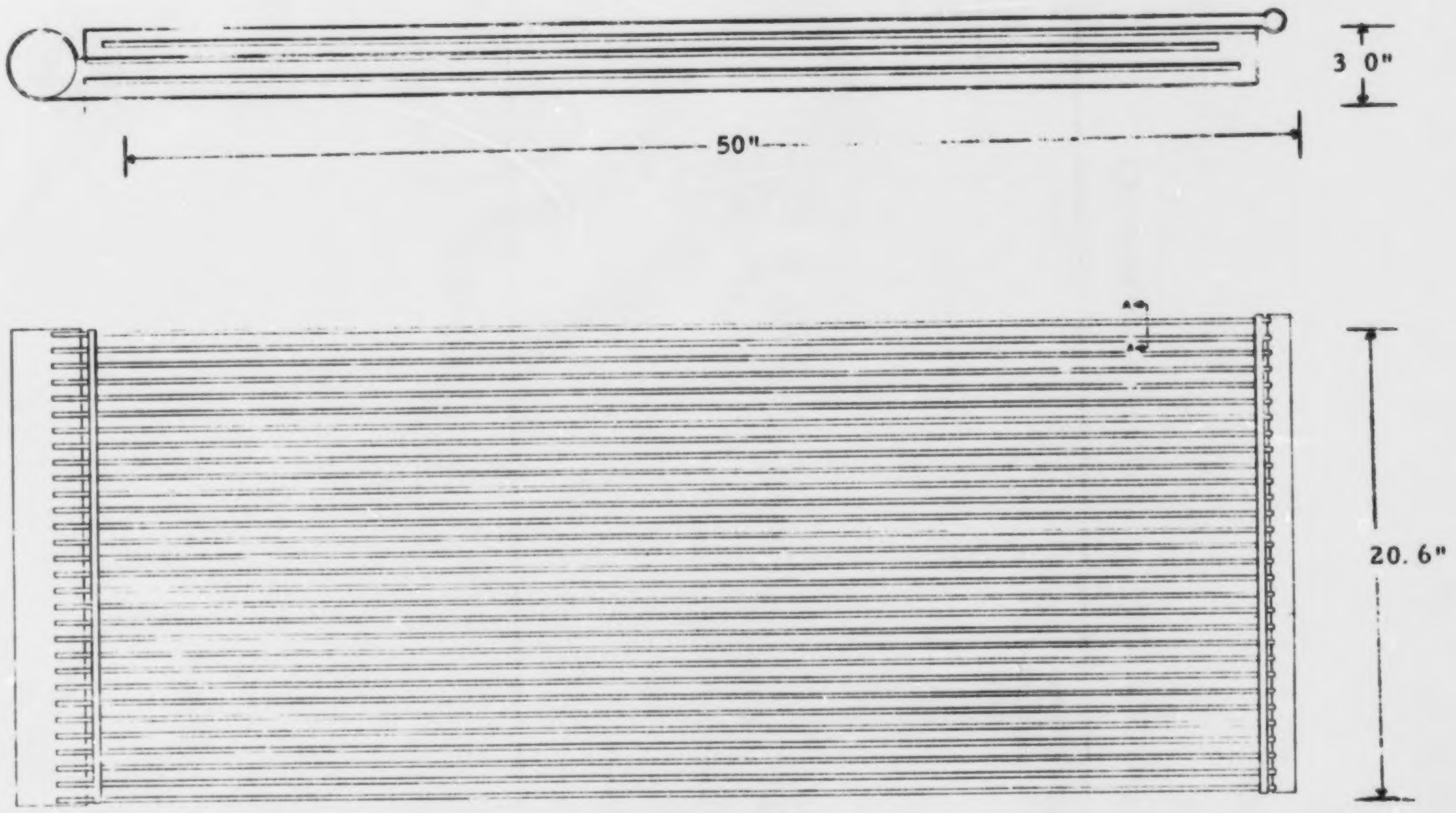
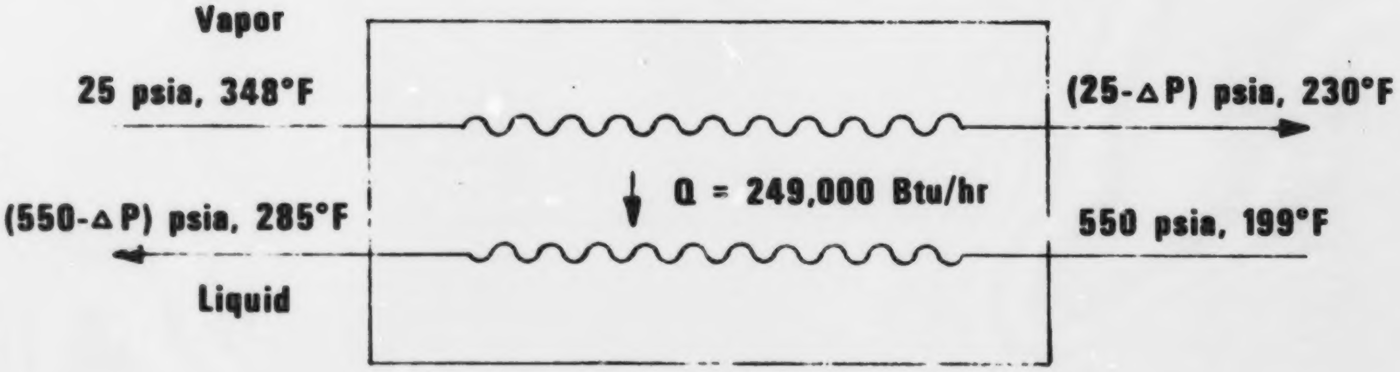


Figure 18. Condenser Design.

TABLE 9

CONDENSER DESIGN POINT CHARACTERISTICS

Heat Rejection Rate (20 °F Superheat Entering 20 °F Subcooling Leaving)	1.25×10^6 Btu/hr
Length	50 in.
Height	20.6 in.
Depth	3.0 in.
Frontal Area	7.15 ft^2
Condenser Inlet Pressure	25 psia
Design Ambient Air Temperature	95 °F
Air Pressure Drop	3.45 in. wc
Ideal Fan Power	7.91 hp
Air Flow Rate	63,000 lb/hr



Liquid Flow Rate = Vapor Flow Rate = 7377 lbs/hr

Figure 19. Regenerator Design Point Requirements.

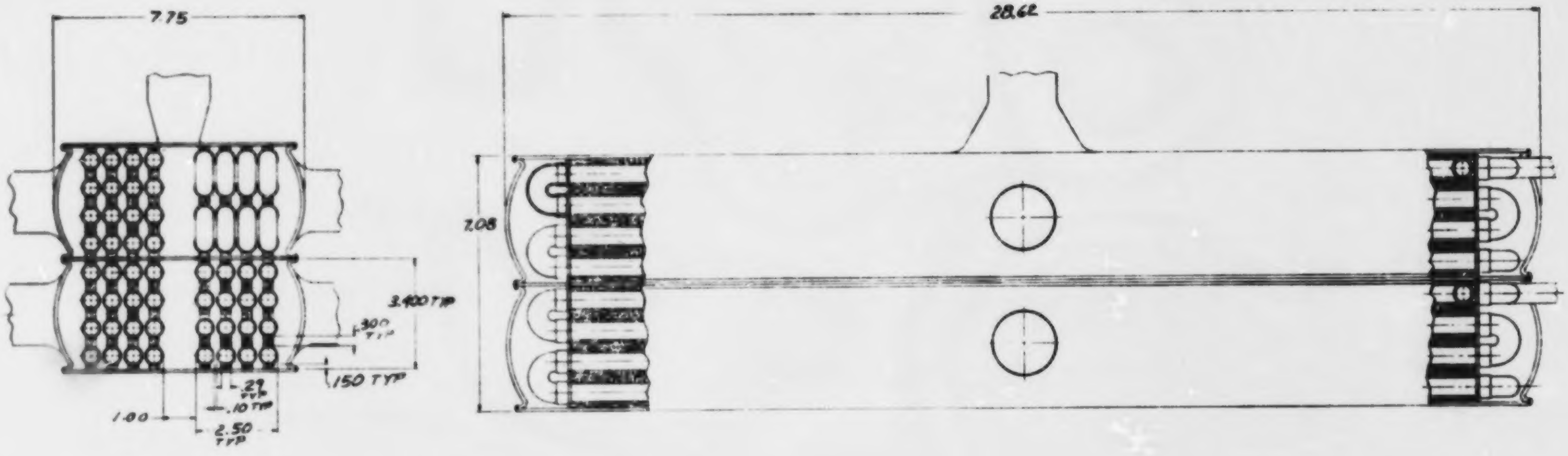


Figure 20. Regenerator Design.

Fluorinol-85 and only minor design changes are expected. The design is based on obtaining a compact regenerator with geometry suitable for packaging directly above the expander in the expander compartment and with low pressure drop on both the liquid and vapor sides. On the vapor side, a brazed ball matrix extended surface with 1/16" ball diameter is used. The exchanger is divided into four parallel liquid circuits; in each circuit, the vapor passes through four separate stages, permitting the exchanger to approach a pure counterflow exchanger.

The regenerator is also designed to function as an oil separator.* The low vapor velocities coupled with the ball matrix should provide very efficient separation of the lubricating oil droplets from the exhaust vapor from the expander. A drain line returns the collected oil back to the expander crankcase.

2.7 Automatic Transmission

An automatic transmission will be used which permits the main propulsion expander to idle at zero vehicle speed and thereby drive the system and vehicle accessories. The use of a transmission also permits matching very closely the driving characteristics of current automobiles powered by the I/C engine with three-speed automatic transmission. The Dana Corporation, Toledo, Ohio, as part of Phase II of the APCO program, is designing a two-speed slipping clutch transmission based on the type of transmission they develop and produce for off-the-road vehicles.

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*Patent Application Pending

2.8 Controls

The startup and control system is designed to provide completely automatic startup and operation of the system. The functions required of the driver are identical to those now required in current, conventional cars with automatic transmission. The controls are assembled from conventional components with the exception of the burner fuel-air control, which is currently under development at TECO.

The primary control problem in the system is control of the burning rate and of the pumping rate to the monotube vapor generator to maintain boiler outlet pressure and the temperature within specified limits over any type of transient encountered by the system. The control system is based on operating the boiler as close to quasi-steady state as possible over all transients; both burning rate and pumping rate are maintained as closely as possible to the values corresponding to the instantaneous vapor flow rate from the regenerator.

A schematic layout of the principal control components is presented in Figure 21. The feedpump control, used to control the pumping rate to maintain boiler outlet pressure, is simplified by the fact that the organic flow rate at any expander rpm is approximately linear with intake ratio. The feedpump and expander valving are thus directly operated as a unit by the accelerator pedal; a vernier control working from the boiler outlet pressure is used to reduce deviations from the design point and to eliminate any unbalance in the system automatically. A mechanical governor is used to limit the maximum intake ratio as a function of rpm and to govern expander speed at idle.

In order to maintain the burning rate at a value corresponding to the organic flow rate into the boiler, the burner control uses an orifice in the organic line to sense almost instantaneously any changes in the

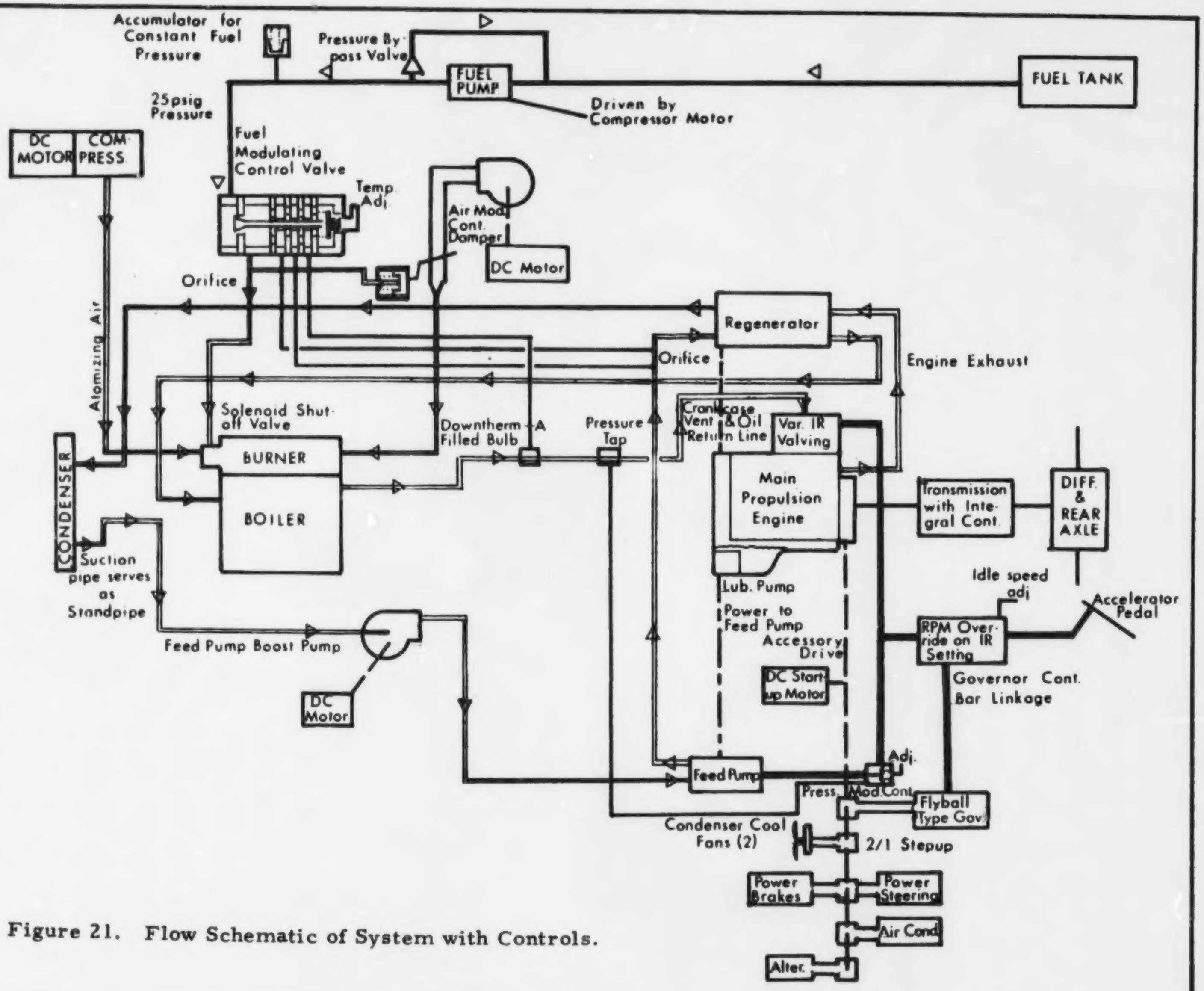

Figure 21. Flow Schematic of System with Controls.

organic flow rate. This signal, along with a similar signal from an orifice in the fuel line, is used with a diaphragm controller to provide the proper fuel and air flow rates. If necessary, the fuel-to-air ratio can easily be varied as a function of turndown to minimize pollutant emissions at any burning rate. A vernier control operating from the boiler outlet temperature is used to reduce deviations from the design point value; at low power levels, when the organic flow is low, the temperature control becomes the primary control on the burner. The fuel control valve cross section is presented in Figure 22 and the air control valve cross section is presented in Figure 23.

3. SYSTEM PERFORMANCE AND PACKAGING

3.1 Performance

The important decisions which have a strong influence on the overall system performance and cost relative to the internal combustion engine are the method of driving accessories at zero vehicle speed and the choice between constant intake ratio expander valving with throttle valve control and variable intake ratio expander valving. With respect to the first decision, two alternatives are feasible. An auxiliary, constant speed expander can be used to drive all accessories, permitting the expander to be coupled directly to the driveshaft (through a simple geat transmission for forward, reverse, neutral, park control). A more complex transmission can also be used, permitting the main propulsion expander to idle at zero vehicle speed so that all accessories can be driven directly by the main propulsion expander; this transmission is still simpler than that required for the I/C engine-driven system.

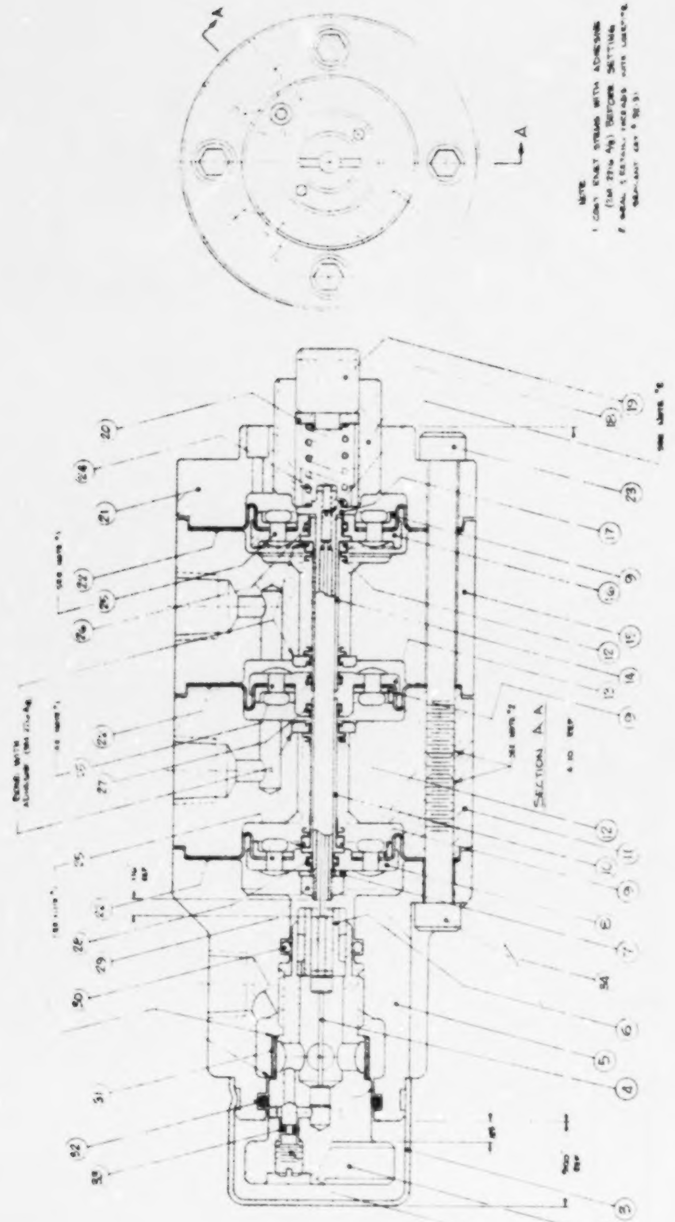

Figure 22. Fuel Control Valve Assembly.

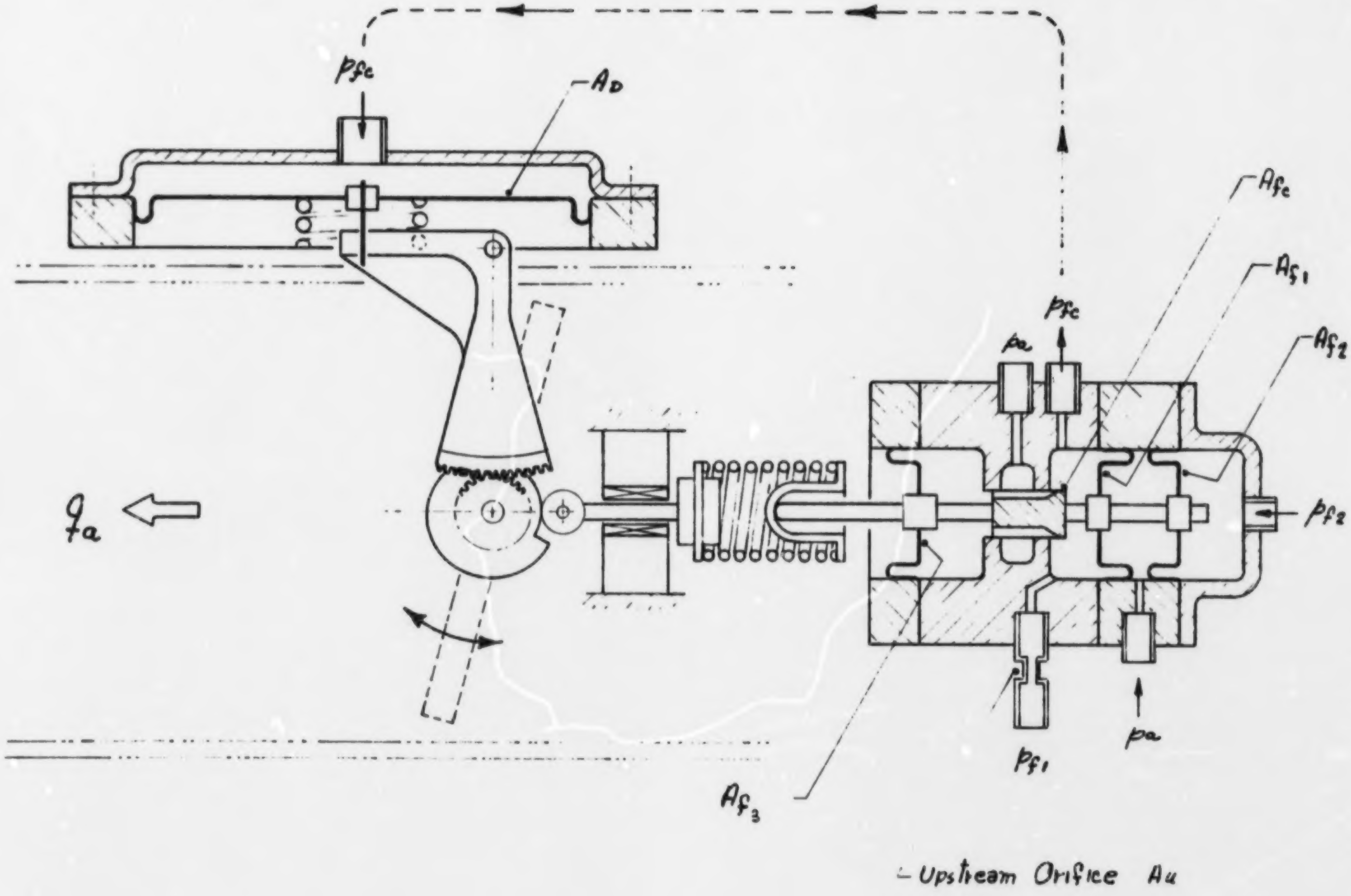

Figure 23. Combustion Air Control.

It has been the conclusion of this study that the use of the main propulsion expander to drive all accessories is preferable. Even though the accessories must be larger when driven by the variable-speed main propulsion expander, the accessory designs are identical and the same number of parts must be processed; very little cost differential exists between the different sized components. Any cost reductions due to smaller accessory components are more than counterbalanced by the requirement for an additional expander of 15 - 20 hp to handle short term accessory peak loads, with governor-throttle valve control. The system with all accessories driven by the main propulsion expander, therefore, seems preferable in terms of simplicity, cost, and packaging; this system has been selected as the optimum approach.

With respect to expander valving, a detailed analysis with computer modeling of the expander and boiler performance for all operating conditions has been carried out, comparing a system with constant intake ratio expander valving and throttle valve control and a system with variable intake ratio expander valving. The results are summarized in the performance maps presented in Figures 24 and 25 for constant IR and variable IR ($IR_{max} = 0.29$) systems, respectively. Comparing these two performance maps for the same boiler and expander sizes, the following conclusions can be made:

a. The system with variable IR valving has a peak efficiency of 18.5% versus 15.0% for the system with constant IR valving. This 20-25% improvement in efficiency (or mpg) occurs over a large power-speed region including the region

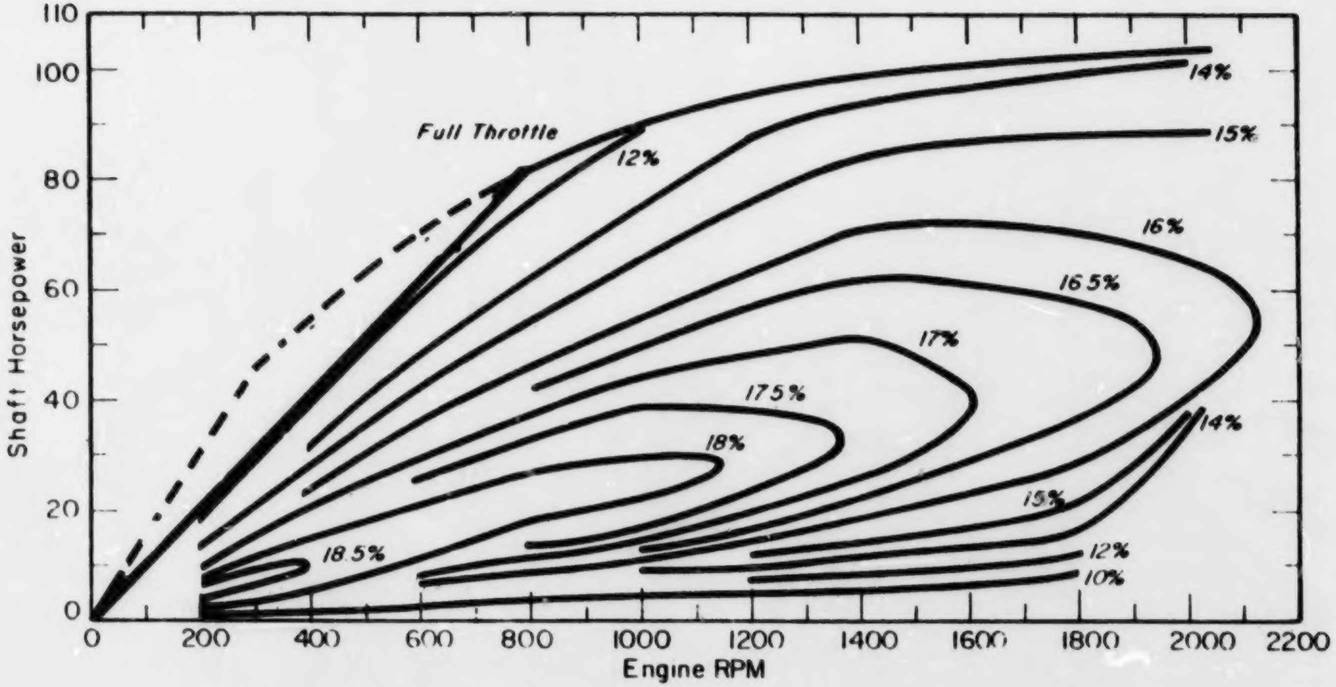

Figure 24. Performance Map with 184 CID Expander and Maximum Intake Ratio of 0. 29.

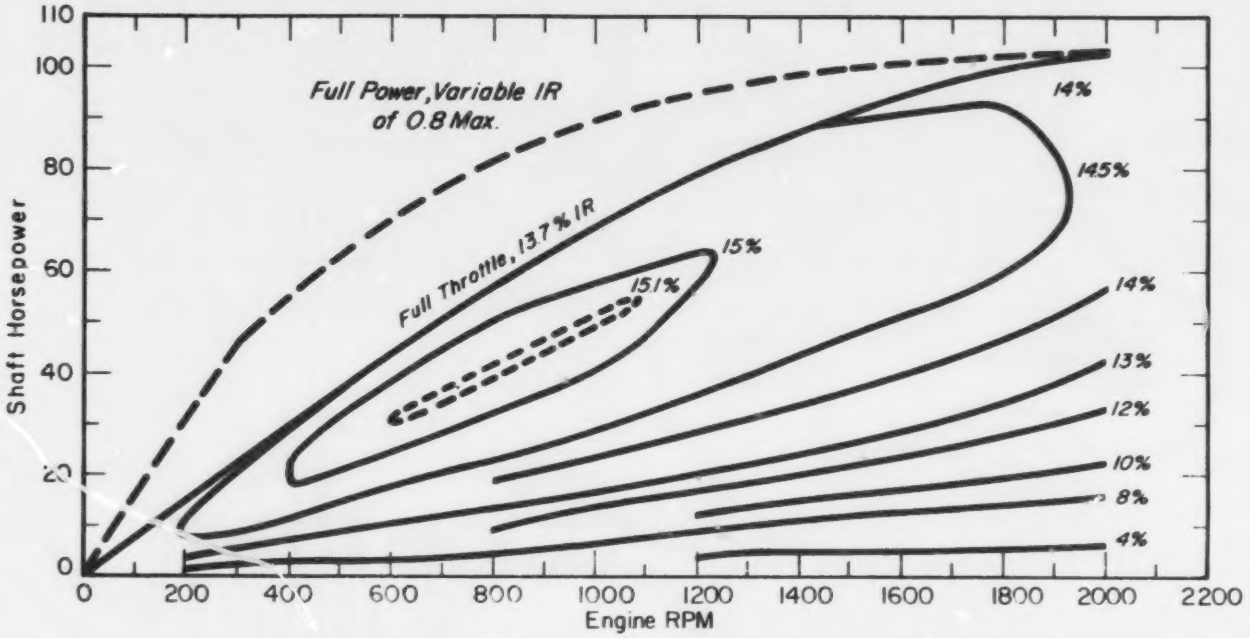

Figure 25. Performance Map with 184 CID Expander and Constant Intake Ratio of 0.137.

of 20 - 40 hp and 600 - 1200 rpm, where the system would operate most of the time.

b. The peak power for the equivalent-sized expander and boiler is much greater over most of the expander speed range for variable IR valving than for constant IR valving. Calculations of 0 - 60 mph wide-open-throttle times indicate a 50% increase for constant IR valving relative to variable IR valving.

For these two reasons, there exists an extremely strong incentive for incorporation of variable intake ratio valving in the expander; this type of valving is used in the reference expander design presented earlier.

In Figure 25, the dotted line to the left of the plot represents the peak system power if a maximum intake ratio of 0.8 (the maximum practical) rather than 0.29 were used. $(IR)_{max} = 0.29$ was selected as optimum for the following reasons:

a. As evident from Figure 25, very little performance is lost by decreasing $(IR)_{max}$ from 0.8 to 0.29.

b. The required feedpump displacement is decreased by a factor of 2.7.

c. The low vehicle speed condenser rejection rate is reduced, permitting greater utilization of ram air.

d. The wide-open-throttle system efficiency at expander speeds to 800 rpm is increased.

With reference to Figure 25, the part-load performance of the system is an extremely important consideration. Thus, while the design point efficiency, defined by the peak power requirements for acceleration, is 13 7%, the efficiency increases under part-load conditions where an automobile normally operates The increase with the variable IR valving occurs because part-load operation is obtained by reducing the IR below the design point value of 0.137, providing a more efficient expansion in the expander, reduction of the condenser pressure under part-load operation also occurs, again leading to a more efficient cycle. The region of high efficiency (≥ 17%) is broad, that is, a high efficiency is obtained over a broad range of engine power and speed. Thus, while the peak thermal efficiency of the Rankine-cycle system (18.5%) is much less than that of the I/C automotive engine (~ 30%), the average efficiencies for typical consumer driving cycles are much closer. The Ford Motor Company has calculated, using the performance map of Figure 25, with $(IR)_{max} = 0.8$, the fuel economies for different driving conditions of the Rankine-cycle system with single-speed slipping clutch transmission installed in an intermediate size American car, and compared them directly with the fuel economy calculated for the same body with 302-2V engine and three-speed automatic transmission. The results are summarized in Table 10 for both steady speed and dynamic driving cycle operation. For the customer average driving cycle, the mpg for the 302 CID I/C engine is 15.7, versus 12.7 mpg for the Rankine-cycle system. For steady speed, zero grade operation, the Rankine-cycle engine has better fuel economy than the I/C engine up to 50 mph and poorer fuel economy above 50 mph. While calculations have not yet been made, it is expected that use of a two-speed transmission as planned will improve the city fuel economy significantly, resulting in an improved customer average. This change, coupled with a potential for increasing the peak cycle temperature to 600°F,

TABLE 10

VEHICLE ECONOMY PROJECTIONS
THERMO ELECTRON CORPORATION
RANKINE-CYCLE ENGINE

VEHICLE: Intermediate size American car, Wheelbase - 116 in.
TIRES: 7.75 x 14, Rolling Radius - 1.08 ft., Rev/Mile - 778

$(IR)_{max} = 0.8$

	Idling			Fuel Economy Computer Program PB1213							
						Steady Speed mph					
	Speed rpm	Fuel Flow lbs/hr	City mpg	Suburban mpg	Customer Average mpg	30 mpg	40 mpg	50 mpg	60 mpg	70 mpg	30-70 mph Average mpg
Ford Production Engines (1969 Model)											
302-2V 3-Speed Automatic	500	3.75	13.3	18.0	15.7	27.3	23.3	20.4	18.0	16.2	21.0
250-1V 3-Speed Automatic	600	2.68	12.9	19.6	16.3	27.0	25.4	22.9	20.1	17.6	22.6
Rankine-Cycle Engines											
182-CID Clutch Drive Single Speed	300	2.00	9.6	15.8	12.7	33.1	25.1	20.4	16.7	13.4	21.7
182-CID Torque Converter, 1.88 Ratio Speed-Up Gear	300	2.00	10.1	15.0	12.5	32.4	24.6	19.7	16.0	13.0	21.2
182-CID Torque Converter 2.77 Ratio Speed-Up Gear	300	2.00	10.3	15.2	12.8	32.3	24.8	19.5	15.7	12.9	21.0

could ultimately result in better fuel economy than the 1969 302-2V I/C engine. It should also be noted that the fuel economy of the I/C engines decreases year by year as additional pollution controls are required.

The acceleration performance of the vehicle with a given powerplant is strongly dependent on the transmission used.

Performance characteristics of the Rankine-cycle system in an intermediate size American car are presented in Figures 26 through 29 and Tables 11 and 12 for a torque converter transmission and single-speed clutch transmission with $IR_{max} = 0.8$ and for single and two-speed clutch transmissions with $IR_{max} = 0.29$. These calculations were prepared by the Ford Motor Company and the Dana Corporation, using existing computer programs, for the Rankine-cycle system with different types of transmissions and for the Ford production 302-2V engine with three speed transmission. The basic input to these calculations was the performance maps presented in Figures 24 and 25.

The most important conclusions from these calculations are:

a. The Rankine-cycle system with 184 CID expander (thiophene working fluid) should be capable of providing 0 - 60 mph acceleration times of less than 15.0 seconds, taken as the criterion for acceptable performance. The acceleration performance is fairly dependent on the type of transmission used. A maximum level grade vehicle speed of 90 - 100 mph should be attainable, irrespective of type of transmission used.

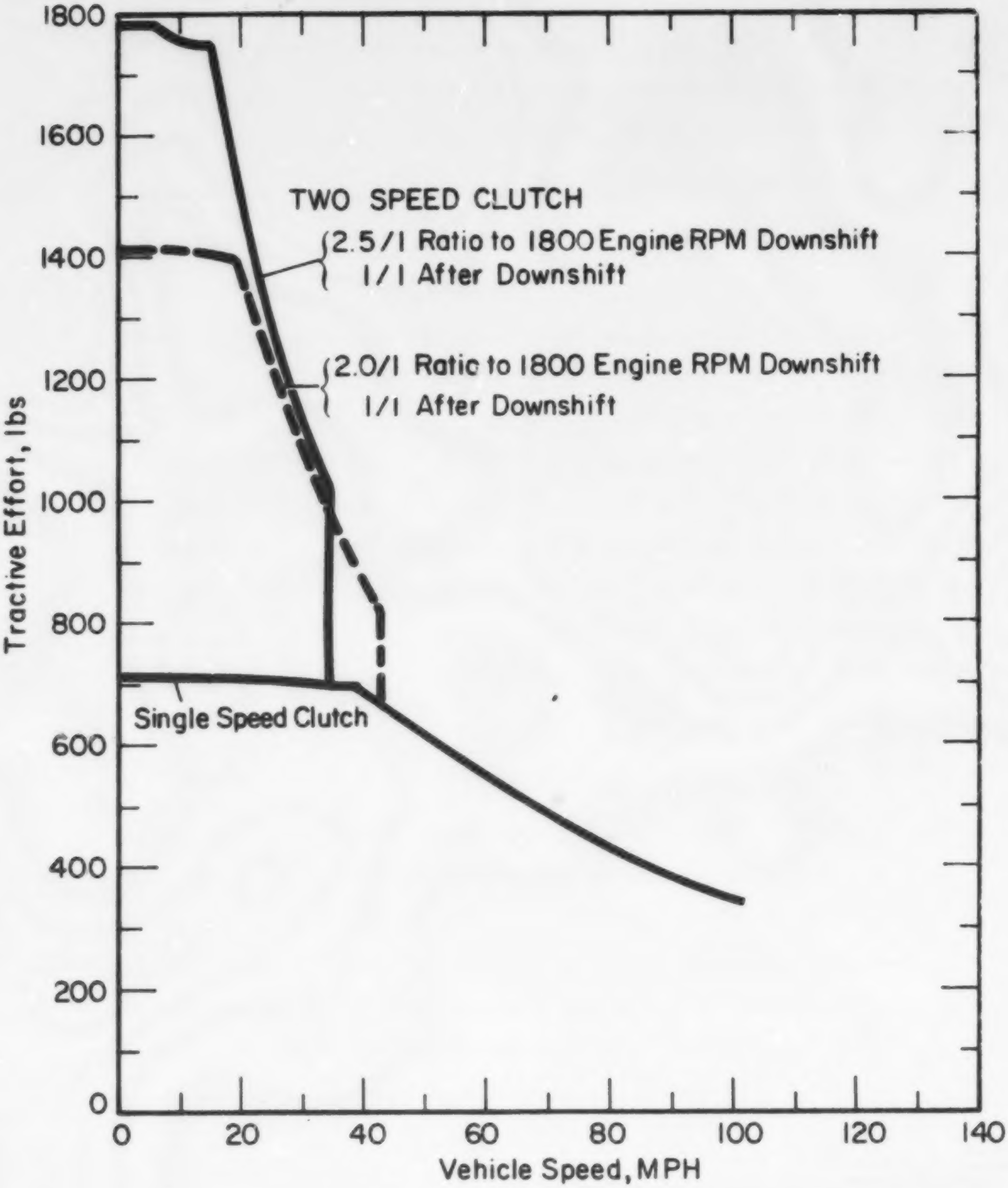

Figure 26. Comparison of Tractive Effort for Single and Two Speed Clutch Transmission.

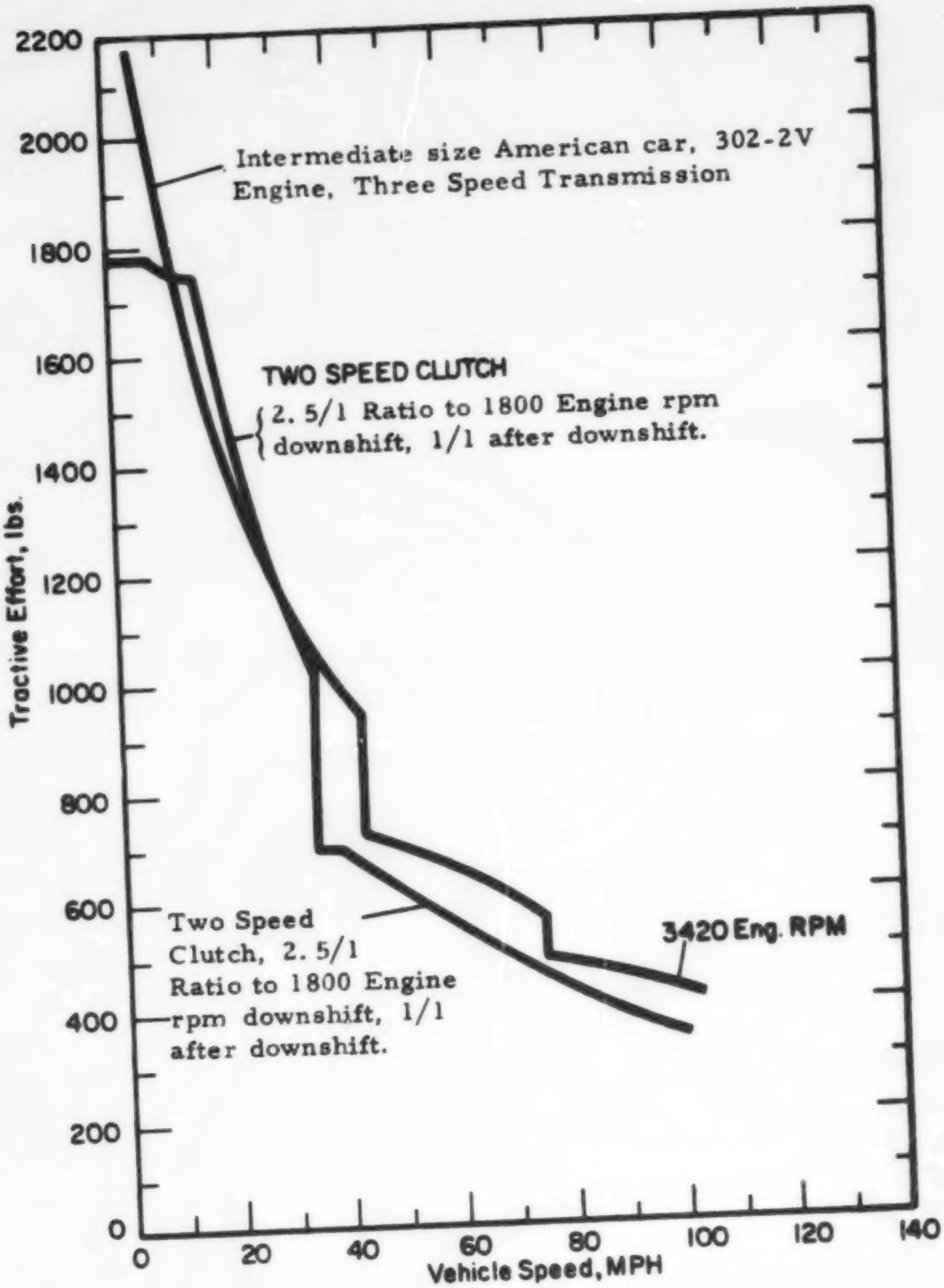

Figure 27. Comparisons of Tractive Effort for an Intermediate Size American car, powered by 302-2V Engine and by Rankine-Cycle System with 104 CID Expander, $IR_{max} = 0.29$, with Dana Two Speed Clutch Transmission.

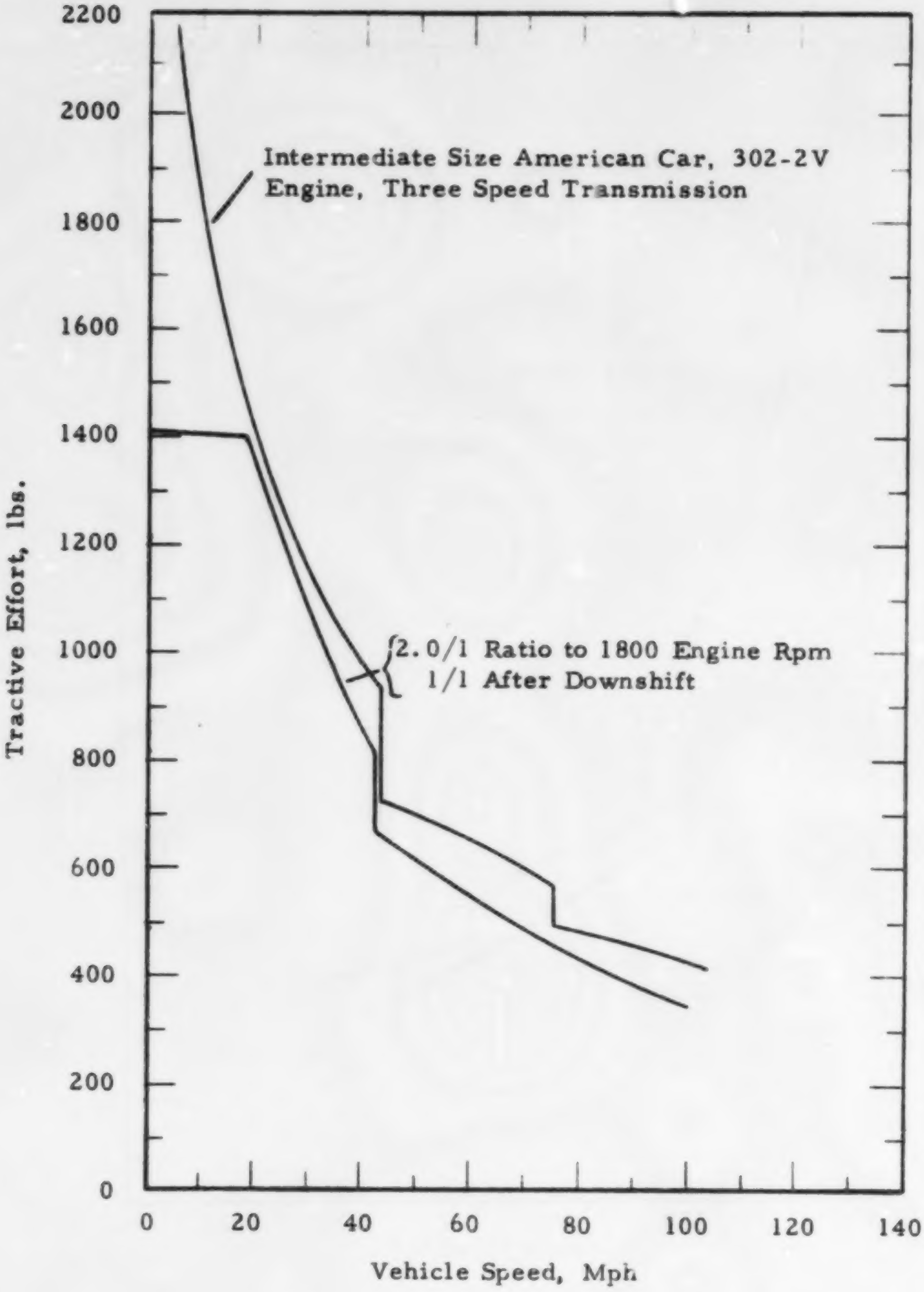

Figure 28. Comparison of Tractive Effort for an Intermediate Size American Car powered by 302-2V Engine and by Rankine Cycle system with 184 CID Expander, $IR_{max} = 0.29$, with Dana two-speed clutch transmission.

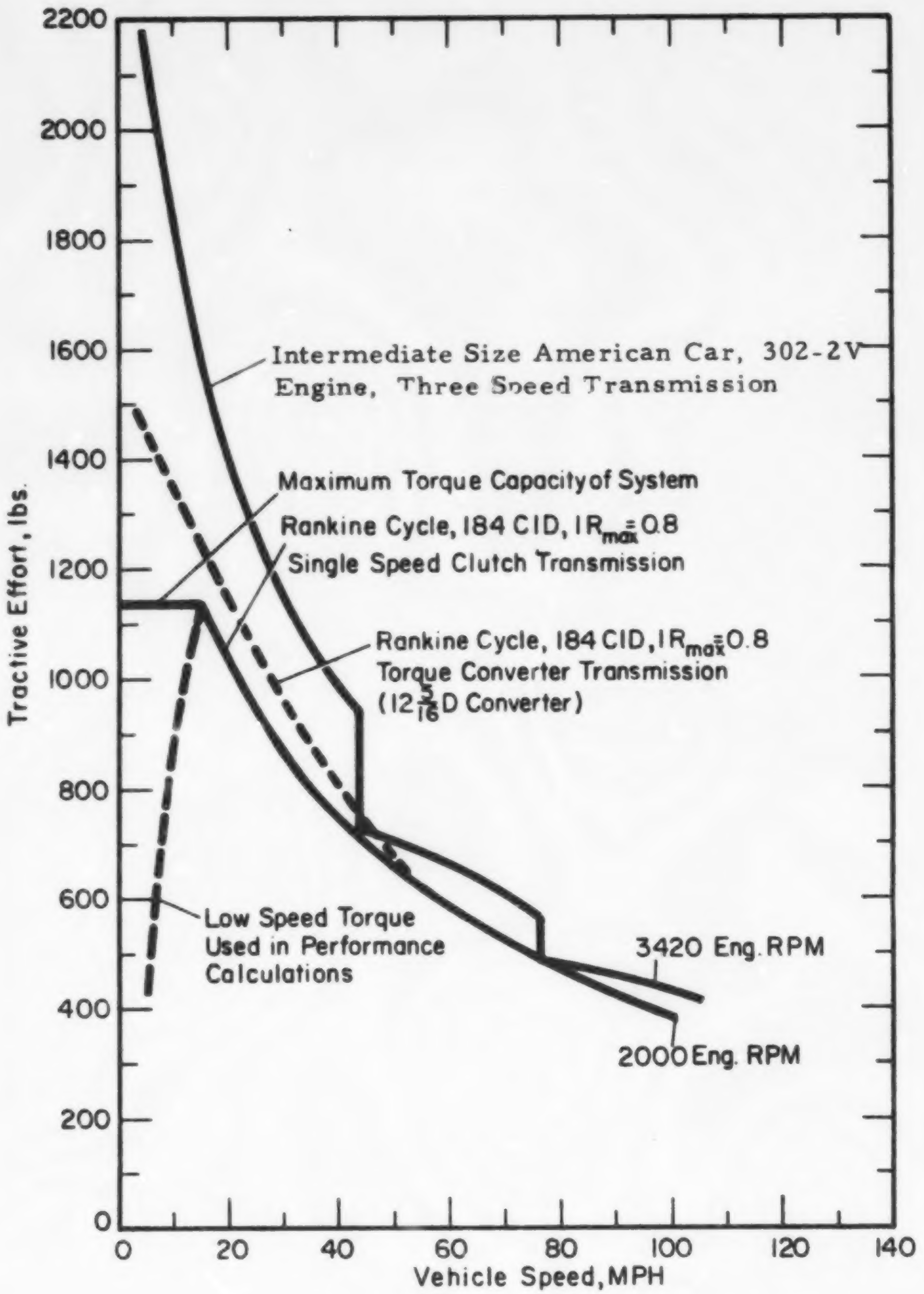

Figure 29. Comparison of Tractive Effort for an Intermediate Size American Car Powered by 302-2V Engine and by Rankine-Cycle System with 184 CID Expander, $IR_{max} = 0.8$, with Single Speed Clutch Transmission or with Torque Converter Transmission.

TABLE 11

VEHICLE PERFORMANCE PROJECTIONS
THERMO ELECTRON CORPORATION
RANKINE CYCLE ENGINE

VEHICLE: Intermediate Size American Car, Wheelbase - 116 in.

TIRES: 7.75 x 14, Rolling Radius - 1.08 ft., Rev/Mile - 778

$(IR)_{max} = 0.8$

	Transmission Gear Ratio	Axle Ratio	N/V	Car Weight lbs.	Performance - Computer Program PB1111				
					0 - 4 Sec. ft.	0 - 10 Sec. ft.	0 - 60 mph sec.	0 - 1/4 Mile sec.	Passing at 50 mph sec.
Ford Production Engines									
302-2V 3-Speed Automatic 11-1/4 Dia. Converter	2.46, 1.46, 1.00	2.79	36.2	3539	89.9	469.2	11.9	18.8	9.72
250-1V 3-Speed Automatic 11-1/4 Dia. Converter	2.46, 1.46, 1.00	2.79	36.2	3522	78.2	405.7	15.6	20.6	11.59
Rankine Cycle Engines									
182-CID Clutch Drive, Single Speed	None	1.64	21.3	3539	70.0	407.9	14.2	20.3	10.30
200-CID Clutch Drive, Single Speed	None	1.64	21.3	3539	77.2	442.0	12.7	19.5	9.74
220-CID Clutch Drive, Single Speed	None	1.64	21.3	3539	86.0	480.4	11.4	18.7	9.22
240-CID Clutch Drive, Single Speed	None	1.64	21.3	3539	94.4	516.7	10.3	18.0	8.79
182-CID 1.88 Ratio Speed-Up Gear 12-5/16 Dia. Converter	None	3.06	21.3	3539	74.2	412.2	14.4	20.1	10.97
182-CID 2.77 Ratio Speed-Up Gear 12-5/16 Dia. Converter	None	4.50	21.3	3539	79.8	414.6	14.3	20.1	10.85

TABLE 12

VEHICLE PERFORMANCE PROJECTIONS.

$IR_{max} = 0.29$

54 ft # Subtracted from Full Torque Curve at All Speeds for Accessories

Transmission	Gear Ratio	Engine Downshift Speed	0-60 mph Acceleration Time Seconds	Gradability
Single Speed	1/1	-	19.6	17.0%
Two Speed	2.5/1 1/1	1800	14.4	48.9%
Two Speed	2.5/1 1/1	1800	14.5	42.5%
Two Speed	2.00/1 1/1	1800	14.6	37.2%
Two Speed	1.75/1 1/1	1800	14.9	31.9%
Two Speed	1.50/1 1/1	1800	15.5	26.7%

b. The Rankine-cycle system with two-speed clutch transmission and $IR_{max} = 0.29$ provides a close approximation to the tractive effort delivered by the 302-2V internal combustion engine with three-speed automatic transmission.

c. The Rankine-cycle system with two-speed clutch transmission and $IR_{max} = 0.29$ provides a gradability of 49% with 54 ft-lbs of torque subtracted for driving accessories.

3.2 System Weight and Estimated Cost

In Table 13, the weight of the Rankine-cycle system reference design based on thiophene (957 lbs total) is compared with that of the 302 CID engine with three-speed transmission (806 lbs total). Substantial weight reductions may be expected in the Rankine-cycle system with development. For example, use of an all-aluminum condenser would reduce the weight of this component by approximately 50 lbs. Use of Fluorinal-85 will also permit reducing the expander displacement from 184 in^3 to 107 in^3, with a resulting reduction in the expander weight.

Using the conceptual design drawings, cost estimators from the Ford Motor Company have prepared a preliminary large-volume production manufacturing cost estimate for the system. The cost was quoted relative to the 1970 302-2V with the result:

$$\frac{\text{Cost, Rankine Cycle System}}{\text{Cost, I/C System}} = 1.55^{+0.31}_{-0.15}$$

The system manufacturing cost estimate will be updated as the development continues.

TABLE 13

TABULATION OF TOTAL SYSTEM WEIGHT AND COMPARISON WITH 302-2V INTERNAL COMBUSTION SYSTEM WITH 3-SPEED TRANSMISSION

	Rankine Reference Design	302 in^3 V-8 with 3-speed Automatic
Expander Assembly	220	
Feedpump	45	
Expander Subsystem	265 lbs	479 lbs
Transmission	135 lbs	159 lbs
Burner-Boiler	273 lbs	
Regenerator	54 lbs	
Condenser	115 lbs	
Radiator		54 lbs
Controls, Fans, Accessory Drives and other Miscellaneous Components	75 lbs	114 lbs
Total	957 lbs	806 lbs

3.3 System Packaging

For the conceptual design, the system was packaged in the engine compartment of an intermediate size American car; a full-size, complete mockup of the system was constructed in the engine compartment of this car with excellent results. In Figure 30, a photograph of the mockup is presented and in Figure 31, sketches illustrating the expander transmission, burner-boiler, regenerator, and condenser locations in the system are illustrated. The only change required in the engine compartment in packaging the system was modification of the frame and fender panels at the very front of the car to facilitate placement of the condenser.

3.4 Emission Projections for Rankine-Cycle Automotive Propulsion System

Using the burner emission levels obtained from the burner developed at Thermo Electron Corporation for a 5 hp Rankine-cycle currently on test, projections have been made of the emission level of unburned hydrocarbons, CO, and NO from a Rankine-cycle automotive propulsion system using 10 mpg fuel economy; the results are presented in Table 14, and compared with projected Federal standards for 1975 and 1980;* presented also are results for an uncontrolled I/C engine and the IIEC targets for emission control for the I/C engine. The Rankine-cycle system emission levels are lower than the projected 1980 standards by a factor of 5 for unburned hydrocarbons, by a factor of 13 for carbon monoxide emission, and by a factor of 1.6 for NO emission.

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* The projected Federal standards in Table 14 are those existing prior to the passage of the Clean Air Legislation in the U. S.

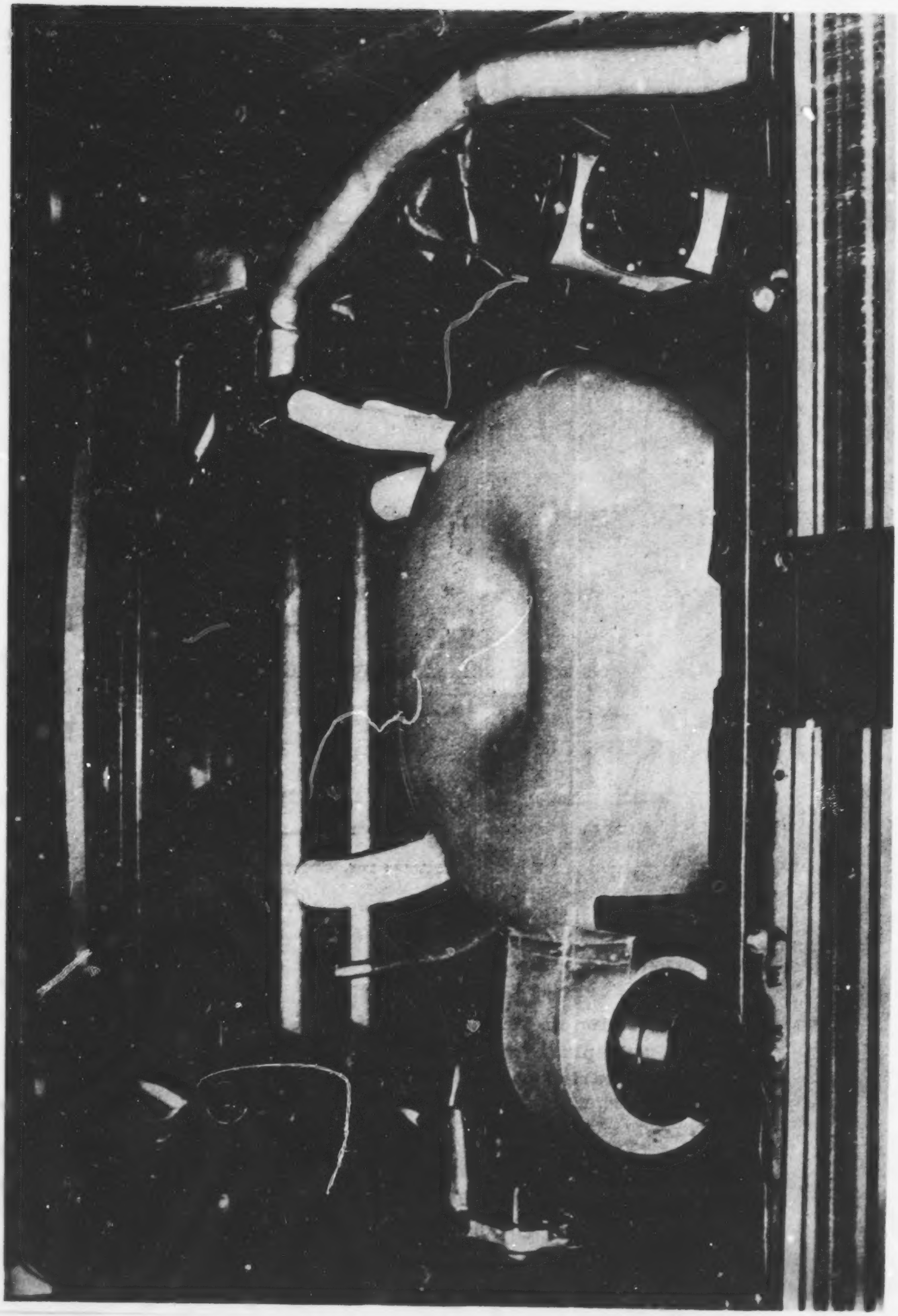

Figure 30. Photograph of Rankine-Cycle System Mockup In Intermediate Size American Car Engine Compartment.

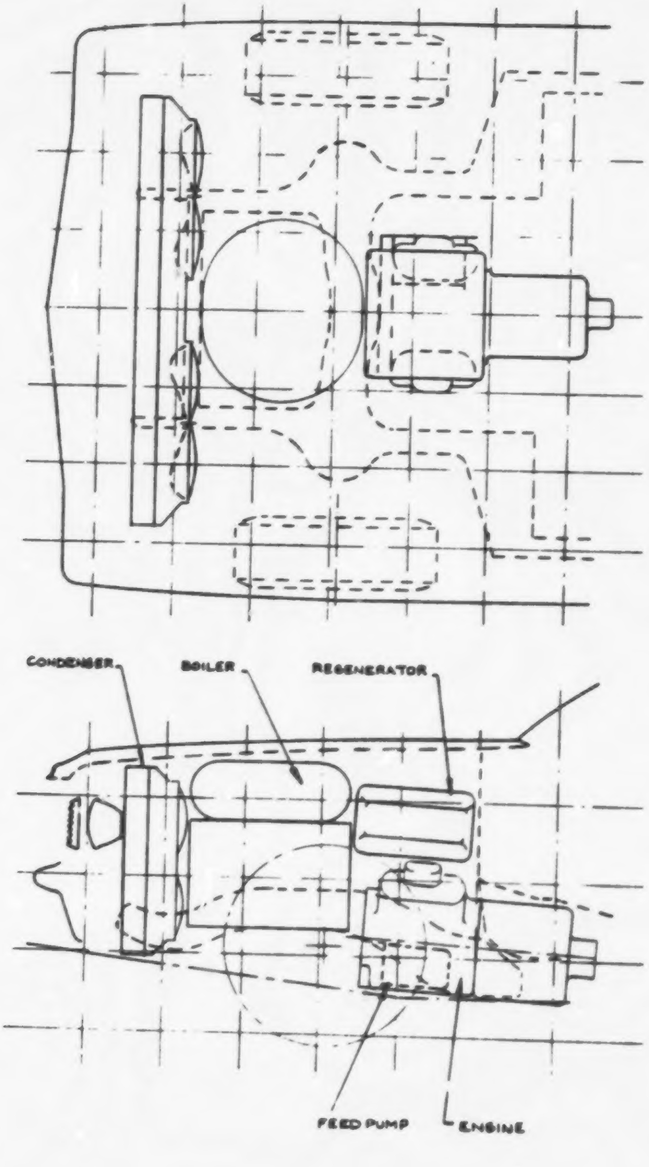

Figure 31. Position of Major Components for 100 hp Rankine-Cycle Powerplant in an Intermediate Size American Car Engine Compartment.

TABLE 14

EMISSION COMPARISON OF RANKINE CYCLE WITH INTERNAL COMBUSTION ENGINE

System		Projected 1975/1976 Standards	Uncontrolled I. C. Engine	IIEC Targets for I. C. Engine	TECO Projections for Rankine-Cycle Propulsion System
Emission	HC	0.46	5.5	0.82	0.05
Level	CO	4.7	24	7.1	0.35
gms/mi	NO	0.4	5.5	0.68	0.25

3.5 Major Conclusions

a. The Rankine-cycle system offers a very strong potential for minimum emissions of particulates, unburned hydrocarbons, nitric oxide, and carbon monoxide.

b. The use of Fluorinol-85 as a working fluid, with a moderate maximum cycle temperature of 550°F, permits a significant cost reduction relative to the equivalent steam system. This reduction may permit the Rankine-cycle system to be competitive costwise with the equivalent I/C system, particularly since the stricter emission level standards will require significant cost increases in the I/C system.

c. An organic-based working fluid, Rankine-cycle system approximately equivalent in performance to a 302 CID I/C engine with 3-speed automatic transmission can be packaged in current automotive engine compartments with only minor modifications required in the sheet metal and frame.

d. The system development and design is sufficiently advanced to insure a high probability of meeting the design goals and performance estimates presented in this report.

Chapter VII

STIRLING ENGINE ACTIVITIES AT UNITED STIRLING (SWEDEN)

by

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Introduction

The development of Stirling engines in Sweden is taken care of by KB United Stirling (Sweden) AB and Company. The main object of the company is the development and adaptation for production of Stirling engines. At present, about 70 engineers are engaged in the Swedish Stirling engine project.

General information on the Stirling engine principles, its properties and potentials is given in a brochure, copies of which are available at the conference. The most important characteristics of this engine are its very clean exhausts and silent operation.

A major part of United Stirling's resources is being used for the development of a 200 hp four cylinder in-line engine, suitable for a number of applications, but especially for operation in areas where the environmental properties are of great importance. Generally speaking, this engine will have outer dimensions allowing the exchange of most existing diesel engines in city buses, mining vehicles, boats and stationary installations.

The 4-615 Engine Program

The 200 hp engine to be produced by United Stirling is named 4-615 (four cylinders, 615 cm^3 of swept volume per cylinder). A development program is in progress, starting with prototype 4-615A engines running in laboratories during 1971. The manufacturing and testing of a number of gradually more advanced prototype engines is scheduled for the following years. Series deliveries are planned to commence in 1976, preceded by earlier pilot production.

A very important background for this development is an extensive use of value analysis technique in order to approach a production version with minimum outer dimensions and manufacturing cost.

The consecutive versions will be gradually more simplified and adapted to series production methods. Further increased realiability and ease of maintenance are other major objects of work. The engine principle, with heat generation and working cycle separated from each other, facilitates independent parallel development of components, e.g., burners, preheaters, control components, etc.

The 4-615 engine is a four cylinder, displacer type engine, with rhombic drive mechanism. Nominal power output is 147 kW (200 hp) at 2,400 rpm at the main output shaft. When choosing design parameters, efficiency has been given a high priority, which, for the rather conservative values of heater tube temperature and mean cycle pressure, has resulted in a rather slow running engine with a relatively large swept volume. Thus the crankshafts turn at 1,550 rpm at nominal power output, and a speed increaser is used to adapt the output shaft speed to existing automotive transmission.

One important result of value analysis work applied to the 4-615 engine is a considerably reduced size, as compared to the present state of art. The estimated dry weight of this engine is 900 -1,000 kg (2,000 - 2,200 pounds).

A brief preliminary specification of this engine is as follows:

Rated gross power	200 hp at 2,400 rpm
Maximum gross torque	80 kpm (580 lbft) at 700 rpm
Maximum brake efficiency in vehicle application	37%
Installation dimensions	
Length	1,200 mm (48 inches)
Width	500 mm (22 inches)
Height	1,100 mm (44 inches)
Fuels	Diesel oil, kerosene
Optional fuels	LPG, LNG

Traction Application Projects

The installation and testing of engines in vehicles has always been considered a very important part of engine development. This is particularly true in the case of Stirling engine development, because of the inherent relations between engine performance (power, efficiency) and properties of the external cooling system.

City bus propulsion is considered one of the most attractive fields of application for United Stirling's engine type 4-615. To make possible a near-future commercial realization of Stirling engine powered buses, a two-phase vehicle development project has been started.

For basic test and evaluation, a Philips type 4-235 engine will, as Phase I, be installed in a test vehicle during 1971. In this phase, theoretical calculations and considerations concerning vehicle performance will be verified. A practically useful theory for matching engines, transmissions and cooling systems will be established. Performance is scheduled to be demonstrated during 1972.

Phase II, which is planned to be completed during 1973, consists of the development of a city bus, using a 4-615 Stirling engine prototype. In this phase, bus and propulsion systems will be mutually matched in order to achieve a near-optimum solution with regard to driving characteristics, space utilization, system noise level, passenger compartment heating, etc. The development will be performed in close cooperation with a bus manufacturer. Experience feedback to production engine design is a very important part of this phase.

As a result of this two-phase development project, production of city buses with exceptionally good environmental properties will be made possible. Besides the very low exhaust emission level described earlier in this paper, a remarkably low noise level will be the result. Referring to the standardized measuring distance of 7.5 metres (25 feet) from vehicle center line, a noise level below 70 dB(A) can be expected under most urban operation conditions.

Chapter VIII

NITROGEN OXIDE FORMATION IN THE COMBUSTION CHAMBER OF THE INTERNAL COMBUSTION ENGINE AND ITS SUPPRESSION BY MEASURES FROM COMBUSTION TECHNOLOGY

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Summary

Production and decomposition of nitric oxide in the internal combustion engine was investigated, with consideration of the kinetics of the elementary chemical reactions. A simple mathematical model of nitric oxide formation in the engine was developed. It consists essentially of an appropriate reaction mechanism for the combustion process in the engine and a thermodynamic analysis of the combustion process. Conclusions obtained from this model concept were then tested on a one-cylinder engine using such measures of combustion technology as stratified charge operation to influence the combustion process so that nitric oxide formation is inhibited because of too low temperature, in spite of the presence of oxygen. In this way the combustion is controlled so that it is still possible to oxidize carbon monoxide and hydrocarbons almost completely.

Contents

1. Introduction

2. Model of Nitric Oxide Formation in the Engine
2.1 Reaction Mechanisms for Nitric Oxide Formation
2.2 Model of Flame Propagation and Thermodynamic Analysis of the Combustion Process
2.3 Calculation of the Kinetics of Nitric Oxide Formation

3. Combustion Technology Measures for Suppression of Nitric Oxide

4. Summary

5. Bibliography

1. Introduction

Investigation of non-equilibrium processes of the processes occurring in the combustion chamber of the internal combustion engine has for some time been of major scientific interest. In view of the drastic measures against air pollution by automobile exhaust which have been required by legislators, it is pressingly necessary to find ways to decrease the injurious materials carbon monoxide (CO), nitric oxide (NO) and unburned hydrocarbons (CH) contained in the exhaust gas, in order to attain more complete combustion and thus a cleaner exhaust gas.

In the following report, the thermal formation of nitric oxide within an internal combustion engine is studied theoretically because of its particular importance for environmental protection. Thus a model of automobile nitric oxide formation was developed, and its transferability to a real engine was studied. On the basis of the conclusions obtained from that, it was attempted to control the combustion process through measures from combustion technology so that the NO concentration would remain as small as possible, and also that the oxidation processes necessary for a reduction of the HC and CO would be completed.

We shall not consider here the potential for NO reduction by exhaust gas recirculation or by use of so-called "diluents" such as CO_2, H_2O, He, Ar, N_2, etc; by use of reduction catalysts; by choice of unconventional fuels, or by addition of certain additives to promote combustion. In this respect, see G. H. Meguerian [11].

2. Model of nitric oxide formation in the engine

2.1 Reaction mechanisms for nitric oxide formation

Thermal nitric oxide formation can be represented by the gross reaction

$$N_2 + O_2 \rightleftharpoons 2NO$$

Such a reaction consists of a number of elementary reactions which, considered microscopically, describe the collisions between molecules, radicals, or atoms, through which new elements are formed. The elementary reactions make up the chemical transformations actually going on, while the gross reaction reflects only the total result of these elementary reactions, without considering the intermediate products.

On the basis of a literature study, the thermal nitric oxide formation in combustion processes can be characterized, for example, by the reaction mechanisms shown in Figure 1. Reaction mechanism I was stated recently by H. K. Newhall and S. N. Shahed [1], and mechanism II by L. S. Caretto and co-workers [2]. Equations (2) to (5) of the first mechanism in Figure 1, to be sure, are contained in the reaction scheme suggested by K. Vetter [3] as early as 1945, while Equations (3) to (8) of the second mechanism had been stated by H. K. Newhall and E. S. Starkman [4] as well as by G. A. Lavoie and co-workers [5]. Since both calculation and experiment have shown [6, 7] that the nitrogen monoxide, NO, is the only nitrogen oxide of significance in the engine, those elementary reactions in these mechanisms which contain nitrogen dioxide, NO_2, or nitrous oxide, N_2O, as participants can be neglected, without causing any large error.

W. Bernhardt [8] showed recently that with respect to nitric oxide formation in the engine, the reactions

$$N + O + M \rightleftharpoons NO + M$$

and $$OH + N \rightleftharpoons NO + H$$

are of secondary importance. See also [9]. Thus two equations remain in each of the reaction mechanisms shown in Figure 1. These are well known under the name of the Zeldovich mechanisms [10].

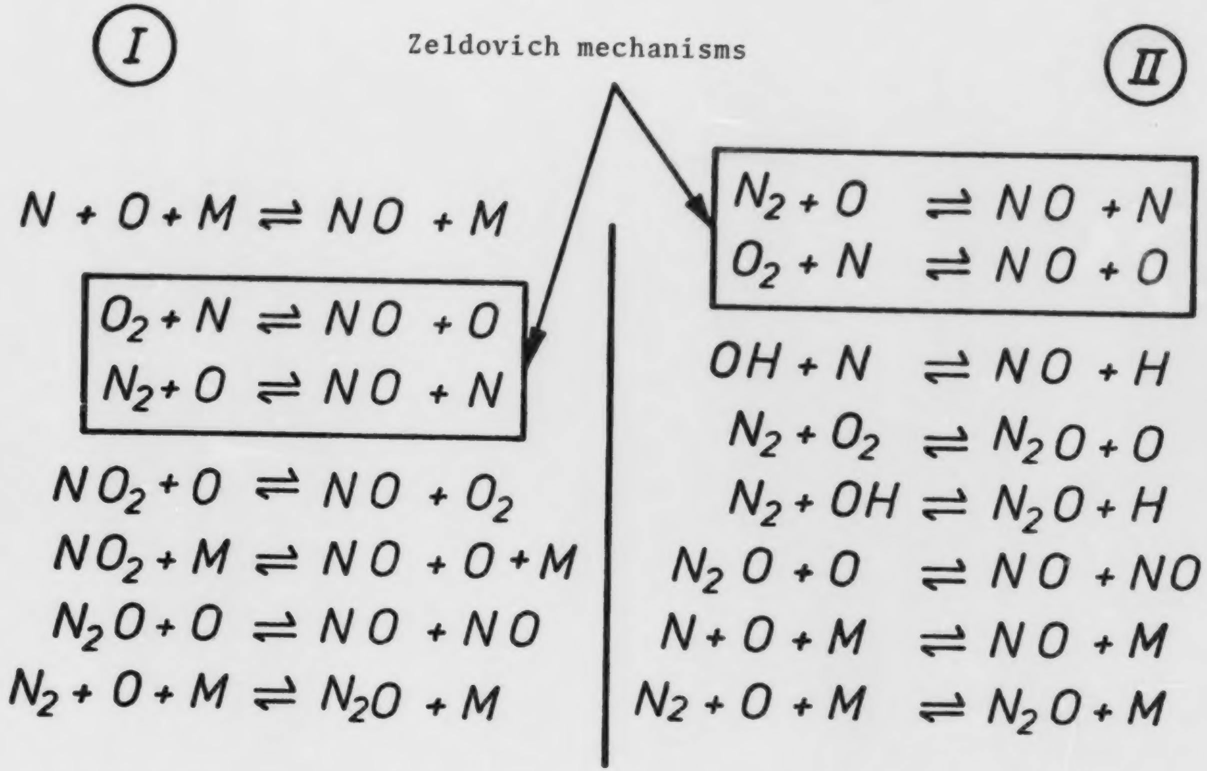
If one compares the velocity constants in the Zeldovich mechanism, using the kinetic reaction data of K. L. Wary and I. D. Teare [12], it appears that the reaction

$$N_2 + O \rightleftharpoons NO + N$$

is clearly the rate-determining reaction for thermal nitric oxide formation because of its high activation energy. See also Figure 4.

Research 2

BE-70-01


I
Zeldovich mechanisms
II
$N + O + M \rightleftharpoons NO + M$
$O_2 + N \rightleftharpoons NO + O$
$N_2 + O \rightleftharpoons NO + N$
$NO_2 + O \rightleftharpoons NO + O_2$
$NO_2 + M \rightleftharpoons NO + O + M$
$N_2O + O \rightleftharpoons NO + NO$
$N_2 + O + M \rightleftharpoons N_2O + M$
$N_2 + O \rightleftharpoons NO + N$
$O_2 + N \rightleftharpoons NO + O$
$OH + N \rightleftharpoons NO + H$
$N_2 + O_2 \rightleftharpoons N_2O + O$
$N_2 + OH \rightleftharpoons N_2O + H$
$N_2O + O \rightleftharpoons NO + NO$
$N + O + M \rightleftharpoons NO + M$
$N_2 + O + M \rightleftharpoons N_2O + M$

The triple collision partner M can be any particle from the reaction volume; it acts only to remove energy.

Figure 1. Reaction mechanisms for the formation of nitric oxide in engines.

2.2 Model of flame propagation and thermodynamic analysis of the combustion process

In the model of flame propagation used in this work it is assumed initially that during the combustion in the combustion chamber there are areas with quite different temperatures. The region already passed through by the flame front contains a burned mixture of high temperature. The region not yet reached by the flame contains fresh gas of relatively low temperature. It is assumed, however, that in both regions it is possible to have intensive mixing and thus a more rapid decay of the temperature differences, and that the energy exchange from the unburned region to the burned region occurs so rapidly that the fresh gas temperature suddenly rises to the higher temperature of the exhaust gas.

So far it has not been possible to establish generally valid analytical relations for the flame advance within an internal combustion engine. It is assumed, however, that the model concept used here sufficiently describes the processes, especially of the actual pressure curve while the combustion process goes on. Along with the pressure, which is taken as constant everywhere in the combustion chamber, the volume V is given as a function of time, from the well-known kinematic equation for piston motion. According to this model concept, the total volume is made up of the volume of the fresh gas and the volume of the burned gas.

The temperature in the unburned region is calculated assuming an isentropic change of state for an ideal gas. This is a crude approximation, because the prerequisite for application of equilibrium thermodynamics is not given. Because of the lesser effect of temperature in the unburned gas on the following calculations, we can proceed in that way here.

The temperature in the burned region can be determined by the time course of conversion of the chemical energy into the internal energy of the combustion gases. For this purpose, of course, a thermodynamic analysis of the processes in the engine must previously have been performed. In this analysis it must be considered that the working fluid changes during the process because of the chemical reaction. Figure 2 shows the result of such an analysis for a VW Type 3 engine (3010 rpm, air ratio $\lambda = 0.93$, full load operation).

Extensive descriptions of the procedure for determining the energy transformations in an internal combustion engine can be found in W. Hinze [13]and W. Bernhardt [14]. For that reason we will not discuss it extensively here. Figure 3 shows the time course of the temperature in the unburned and burned regions of the combustion chamber for the combustion process being studied. The figure also shows the mean temperature in the working gas, as would be given from an adiabatic mixing process. In the temperature determination it was assumed that the state behavior of the reacting substances is given by the equation of state for ideal gases.

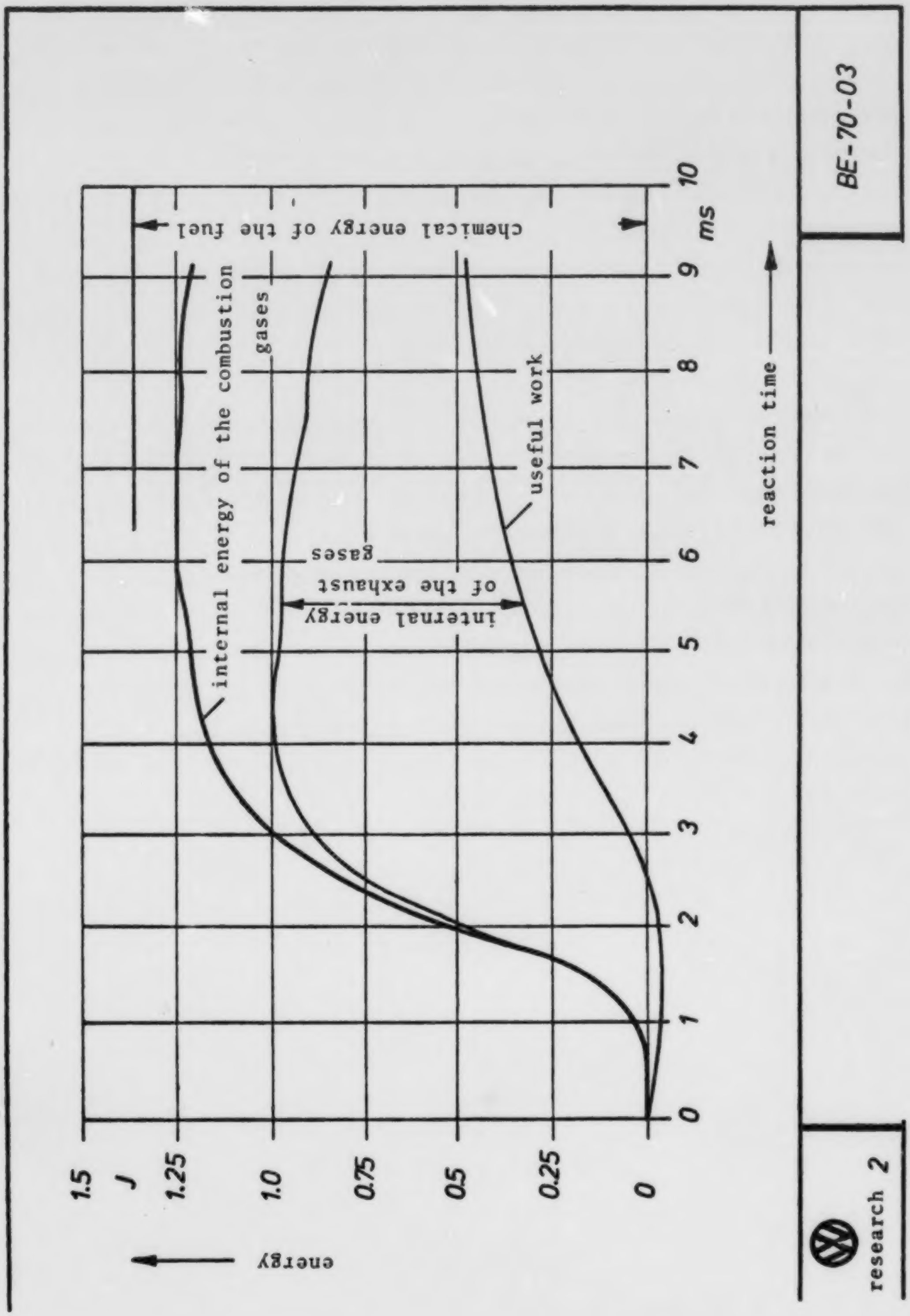


Figure 2. Energy exchange in an internal combustion engine as a function of the reaction time.

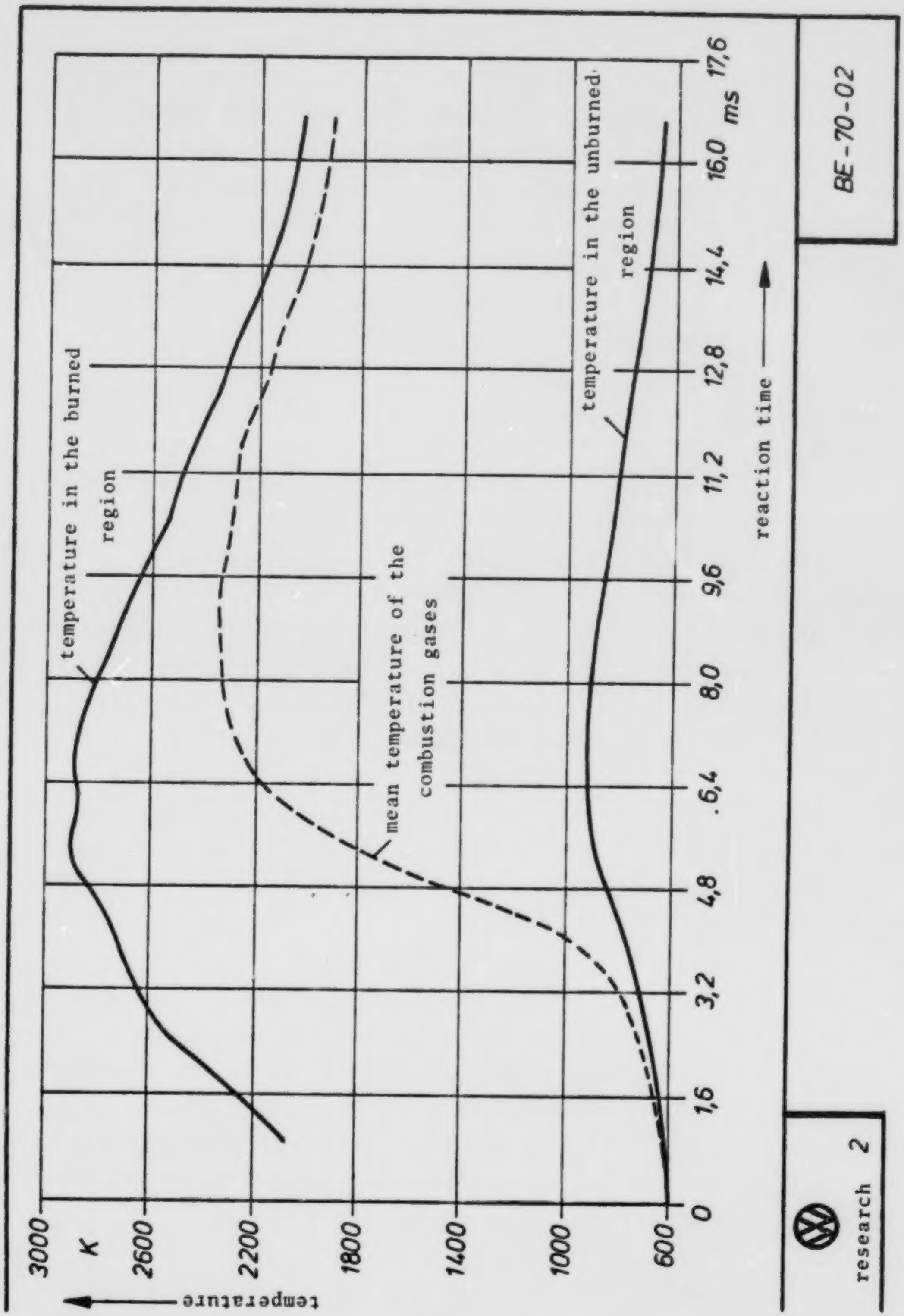


Figure 3. Time course of the temperature in different regions of the combustion chamber.

2.3 Calculation of the kinetics of nitric oxide formation

The model of flame propagation which has been described was used to calculate the kinetics of nitric oxide formation. The temperatures and volumes or densities calculated by means of this model are particularly needed. For the Zeldovich mechanism, the time change of the NO concentration can be written in the form presented in Figure 4 for isochoric processes, in which the density remains constant.

Problems arise in the application of this differential equation, however, because of the determination of the time course of the concentrations of the participants in the reaction, especially of the O atoms. Starting from the known initial concentrations of O_2 and N_2 immediately before the beginning of the combustion process, the concentrations of O_2, O, N_2 and N are calculated approximately for the temperatures in the different regions of the combustion chamber by use of the equilibrium constant, considering the dissociation and the fuel conversion which has already occurred. With the known concentrations σ_{O_2}, σ_{N_2}, σ_O and σ_N , known density, and known temperature, the differential equation in Figure 4 can be solved by means of the method of Runge-Kutta. Thus the NO formation can be followed analytically in the unburned as well as in the burned region of the combustion chamber.

As an example, Figure 5 shows the curve for the NO concentrations in the two regions existing, according to the model concept, with a VW Type 3 engine at 1000 rpm, ignition advanced to 12° before TDC, and an air ratio of $\lambda = 0.93$. The nitric oxide formation is shown graphically, starting at the time of ignition ($t = 0$), during the combustion and expansion processes. The nitric oxide concentration is shown in moles per cm^3.

As was to be expected from studies carried out previously by means of equilibrium theory, the nitric oxide formation in the fresh gas is negligible in comparison to that in the burned

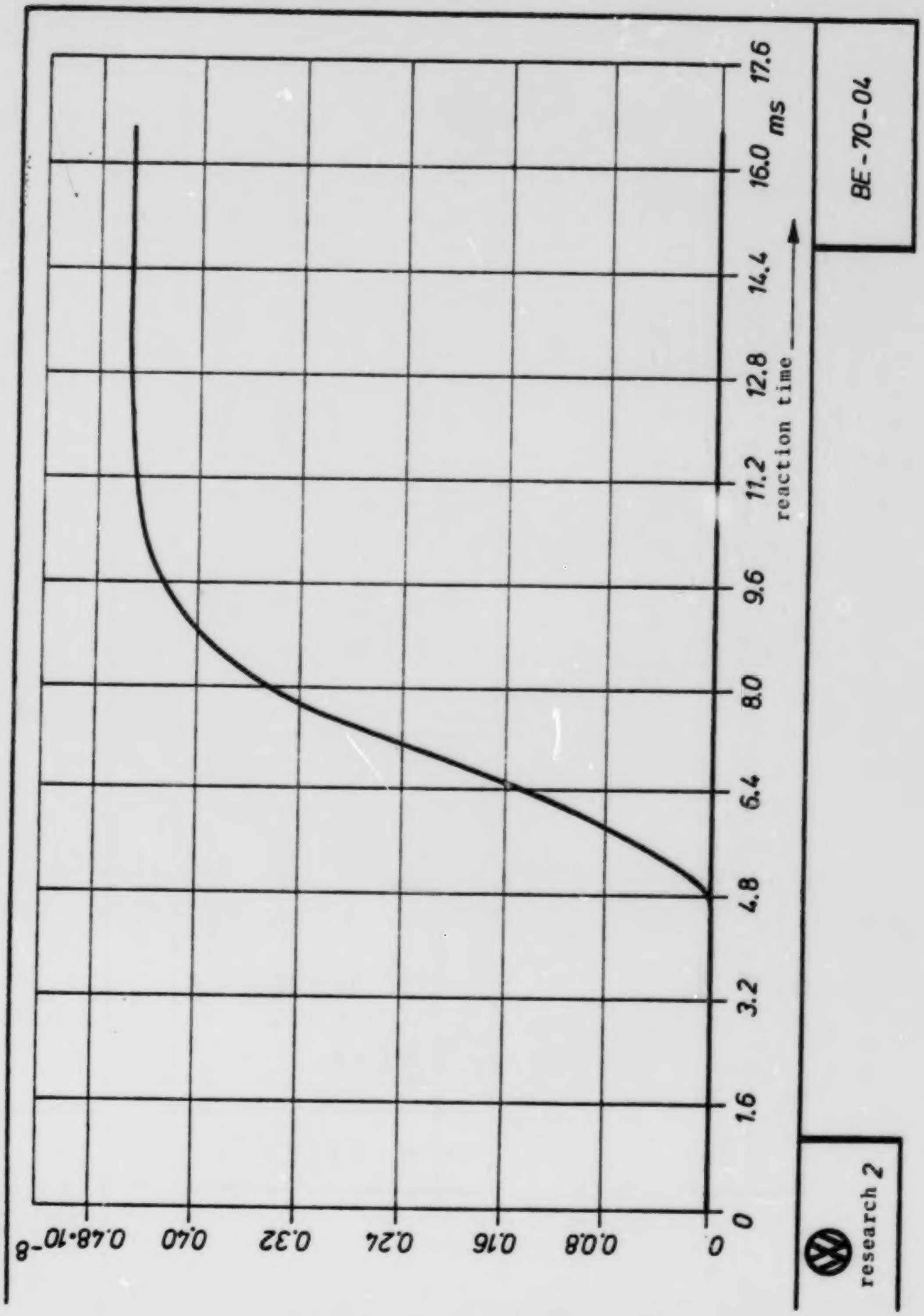


Figure 4. Calculation of the kinetics of NO formation.

time change of the NO concentration for the Zeldovich mechanism:

$$\boxed{\frac{d\sigma_{NO}}{\rho\ dt} = k_3 \sigma_{O_2} \sigma_N - k_4 \sigma_{NO} \sigma_O + k_1 \sigma_{N_2} \sigma_O - k_2 \sigma_{NO} \sigma_N}$$

in which ρ is the density, σ_i is the number of moles of component i per mass, t is the time, and k_j is the velocity constant of reaction j:

j	reaction	velocity constant $\frac{cm^3}{mol \cdot sec}$
1	$N_2 + O \rightarrow NO + N$	$7 \cdot 10^{13} \exp [-75000 / R \cdot T]$
2	$NO + N \rightarrow N_2 + O$	$1,55 \cdot 10^{13}$
3	$O_2 + N \rightarrow NO + O$	$13,3 \cdot 10^{9}\ T \exp [-7080 / R \cdot T]$
4	$NO + O \rightarrow O_2 + N$	$3,2 \cdot 10^{9}\ T \exp [-39100 / R \cdot T]$

T in K , R in cal/[mol · K]

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Figure 5. Time course of the NO concentration.

region because of the lower temperatures in the fresh gas region (< 1000 °K). The nitric oxide concentration in the fresh gas is so slight that it is barely detectable in Figure 5. It is a characteristic of the thermal nitric oxide formation in an Otto engine that it becomes perceptible only quite late, after about a third of the total combustion period has passed. [This applies only for the average nitric oxide concentration, referred to the total combustion volume, which we are considering here.] The maximum of the (average) NO concentration occurs only when two thirds of the combustion period has passed. After the maximum has been reached, there is hardly any detectable change in the nitric oxide concentration during the expansion process. This result is in distinct agreement with the experimental studies of H. K. Newhall and E. S. Starkman [4, 16], who recorded the monochromatic emission of NO by means of a monochromator directly during the expansion, and with the reaction kinetic calculations of H. K. Newhall [17]. On comparison of the engine nitric oxide concentrations in the exhaust gas, as predicted by this model of engine nitric oxide formation, with the values measured for an actual engine, it can be seen that this model is valid only for nearly stoichiometric mexture ratios and for the region $\lambda > 1$. The reason is that in the rich region ($\lambda < 1$) the effect of the fuel is not considered. This has also not been done in other models just recently proposed [18, 19].

According to G. H. Meguerian [11], with a fuel excess in the Zeldovich mechanism, the oxygen and nitrogen radicals react with the unburned or partially burned fuel molecules. This leads to lower nitric oxide concentrations, and the NO values for 15% fuel excess are about 30% below the values obtained for the stoichiometric mixture. See also [19]. In the thin region ($\lambda > 1$) the predicted NO values agree well with those measured in the engine exhaust, because the flame velocity decreases with increasing air ratio, and this effect is considered in the flame propagation model.

3. Combustion Technology Measures for Suppression of Nitric Oxide

In the calculation of the kinetics of nitric oxide formation it appears that temperatures above 2600 °K are necessary for formation of NO in the burned region of the mixture. See Figures 3 and 5. By contrast, the CH and CO oxidation processes go on even at considerably lower temperatures. If it were possible to control the combustion process in the engine so that the temperature of the combustion gas would remain below about 2600 °K, the nitric oxide formation would be inhibited. This would certainly lead to lower NO concentrations in the exhaust gas. Another important conclusion from the model of engine nitric oxide formation is that NO is formed only quite late in large concentrations in the burned mixture. Since the flame front of the burned region has already passed, we speak here of "post-flame reactions".

On the basis of knowledge obtained from the model concept of NO formation, an engine combustion process was tested with a CLR one-cylinder engine in research at the Volkswagenwerke AG. (1) The process appeared to be able to decrease simultaneously the CH and CO concentrations as well as the NO concentration. This amounts essentially to use of a pre-combustion chamber, in which a nearly stoichiometric or even a rich mixture is first ignited and burned. The design of the combustion chamber is such that, after passage through the pre-combustion chamber, the flame front advances into the main combustion chamber. There it meets the load-controlled main mixture at relatively low temperatures. (The main combustion chamber can even contain completely pure air.) In this way the hot combustion gases are rapidly lowered in temperature, so that in spite of excess oxygen only small amounts of NO can be formed. The temperature which results is still high enough, though, to allow rapid oxidation of carbon monoxide and hydrocarbons. A pre-combustion chamber arrangement which is similar in principle was suggested recently by Newhall [20].

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(1) These investigations were performed by Graduate Engineer I. Geiger.

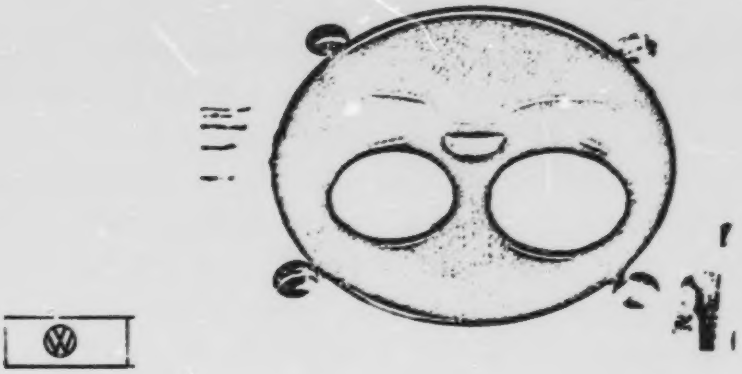

Figure 6. Combustion chamber with pre-combustion chamber.

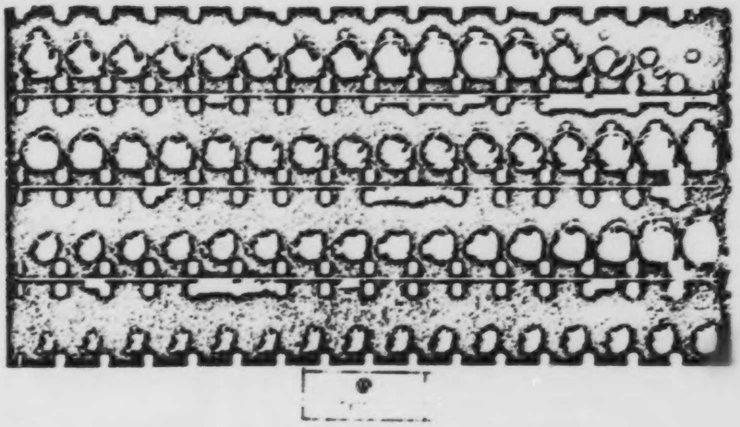

Figure 7. Combustion in a combustion chamber with a pre-combustion chamber.

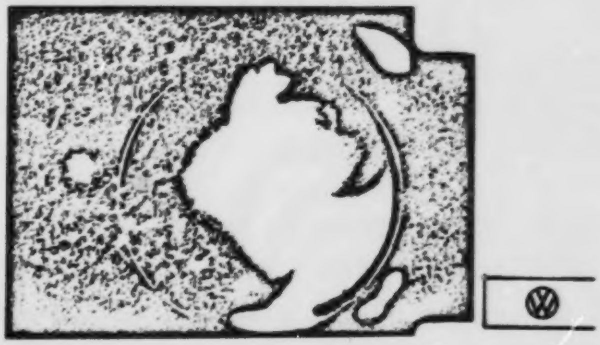

Figure 8. Combustion in a combustion chamber with a pre-combustion chamber.

Figure 6 shows the arrangement of the pre-combustion chamber and the main combustion chamber with a VW Type 3 cylinder head. In these studies, the ratio of pre-combustion chamber volume to main combustion chamber volume was between 1:10 and 3:10. With this pre-combustion chamber arrangement, very slight exhaust gas emissions were measured with a CLR one-cylinder engine at 2000 rpm under full load operation with optimum ignition timing. The emissions were as low as 40 ppm NO, 40 ppm CH (measured as hexane) and 0.1% CO by volume at air ratios of $\lambda = 2$ to $\lambda = 3$.

In conclusion it should be mentioned that the combustion processes in the combustion chamber could be filmed by use of a high-speed camera (6,000 pictures/second) so as to test whether the combustion technology measures applied satisfied the information obtained from the model of engine NO formation. Figure 7 shows 64 pictures from a single combustion process. They were made at 2000 rpm, ignition at 30° before TDC, effective mean pressure $P_{me} = 2.5$ kp/cm^2, an air ratio of $\lambda = 2.5$ in the main combustion chamber and an over-all air ratio $\lambda_{ges} = 1.6$, using the high-speed camera. In this case we measured 170 ppm NO. This is more than an order of magnitude less than in conventional combustion chambers.

Figure 8 shows an enlargement of one photograph from Figure 7.

4. Summary

The nitric oxide concentration in the exhaust gas can be predicted for stoichiometric and fuel-poor mixtures with the simplified model of engine nitric oxide formation described here. It is shown that, along with the velocity constants of the reaction

$$N_2 + O \rightleftharpoons NO + N$$

the temperature in the region passed by the flame front is the principal quantity affecting the prediction.

Based on conclusions from the model, and with basic changes in the previous combustion processes, such as stratified charge operation, exhaust gas emission measurements on a CLR one-cylinder engine with a modified VW Type 3 cylinder head showed NO at 5 to 10% and CH at 10 to 25% of the levels observed with conventional combustion processes. Pre-combustion chamber arrangements, combined with stratified charge operation could prove to be potential solutions for significant reduction of the injurious materials in automobile exhaust gas. In spite of the good results obtained so far, it is doubtful whether this method alone will be suitable to attain the drastic reduction of emissions required by American legislators after 1975, with retention of good specific power and economy.

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Chapter IX

A EUROPEAN CONTRIBUTION TO LOWER VEHICLE EXHAUST EMISSIONS

by

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I do not propose to discuss whether the present limits and proposed future limits for road vehicle exhaust emissions are sensible economically or well founded scientifically. I will only remark in passing that the NO_x regulations will be the most difficult to meet and will have the greatest effect on the economics of operation of motor vehicles. It behoves us to be sure, therefore, before imposing them, that they are really essential medically and environmentally, and that we have our priorities right in relation to other atmospheric pollutants.

In this Note, I have deliberately confined myself to two areas of interest and concern to the automobile engineer

1. The small European automobile and its problems in meeting future US legislative requirements.

2. The contribution which combustion chamber design can make to reducing exhaust emissions from the diesel engine.

The European Automobile and the American Market.

As Consulting Engineers engaged in design, development and research on internal combustion engines, we have been very active in assisting European car manufacturers to meet the U.S. Federal and Californian State exhaust emission limits. Our experience is probably unique in the range of vehicle type we have studied and the variety of approach we have used to meet individual requirements.

To meet the U.S. Regulations up to and including those imposed for the 1971 model year, we have used in the main what has come to be known as the Cleaner Air Package (CAP) approach, involving a tightening of production tolerances and detailed adjustments to the carburettors and ignition settings over the load and speed range, combined with the use of such devices as intake air heaters and manifold air depression limiters. The art is to reach the required exhaust emission levels without unacceptable loss of driveability. This is harder to achieve with the small European car than with the larger and generally more powerful American car. For this reason, a number of European car manufacturers have adopted petrol injection in place of carburation, which, because of the more precise metering and delivery of the fuel over the operating range including the transients, enables the present exhaust emission limits to be attained without sacrifice of driveability; in fact, in most cases, with driveability enhanced. For the 1972 model year, it seems probable that further refinement of the CAP approach, particularly if associated with petrol injection, will enable the lower limits associated with the change of test procedure, to be attained. In some cases, however, it may be necessary to use manifold air oxidation, a device which has been used intermittently since the early days of pollution control, but which, because of the cost of the air pump and the problems posed by its installation and drive, has generally been abandoned wherever possible.

We have been working for some time with our eye on 1975 and Table I summarizes the present position. The first line gives the limits which we thought we were going to have to meet prior to the signing of the Nixon/Muskie bill last December. The second line gives our present target, necessarily somewhat vague in regard to NO_x. Some typical results obtained with small European cars fitted with catalytic afterburners are shown in the second half of the Table. It can be seen that, ignoring NO_x for the moment, the pre-Muskie limits for HC and CO have almost been reached with copper/chromium catalyst and could probably be still further reduced with more development work. With the platinum catalyst, even the limits required by the Nixon/Muskie bill can be met in regard to HC and CO. When the attempt is made to reach the NO_x limits as well, by the use of Exhaust Gas Recirculation (EGR) combined with a platinum catalytic afterburner, the NO_x is certainly reduced, but the HC and CO are then too high. These results were obtained with an unleaded fuel and it should be emphasized that the performance of these catalytic systems in respect of endurance has not yet been fully assessed.

It would seem that it is going to be very difficult, if not impossible, to reach the figures required for 1975 by the use of EGR, without unacceptable loss of driveability. For this reason, we believe that a double catalytic system holds out the best hope for a European car, with a first stage reducing the NO_x and a second stage oxidising the HC and CO. We have no results to report on such a system as yet.

TABLE 1

AUTOMOBILE EXHAUST EMISSIONS
(1972 FEDERAL TEST PROCEDURE)

	HC	CO	NO_x
1975 U.S. FEDERAL LIMITS			
1. Pre Nixon/Muskie	0.5	11.0	0.9
2. Post Nixon/Muskie	0.46	4.8	0.4-0.6
RESULTS WITH SMALL EUROPEAN CARS			
1. Cu/Cr Oxidation Catalyst	0.6	8.5	5.0
2. Pt Oxidation Catalyst	0.3	3.0	5.0
3. Pt Catalyst + EGR	1.7	10.0	1.0

.. ..

Although the US legislative requirements between 1972 and 1975 are not entirely clear, it would appear that we shall have to meet low limits for NO_x in California in 1974. As the figure would not appear to be nearly as low as that proposed for 1975, however, it could probably be met by the use of a small amount of EGR combined with a catalytic afterburner, without too much loss of driveability. The Federal limits for 1975 could probably be met in regard to CO and HC by an oxidation catalyst system. It is going to be very much more difficult to meet the 1975 limits for NO_x, and we in Europe would certainly hope that the application of these limits would be postponed until 1976, as is provided for in the Nixon/Muskie bill.

Exhaust Emissions from the Diesel Engine.

Attempts to lower the exhaust emissions from the spark-ignition engine by changes in combustion chamber design have been disappointingly ineffective. The same is not true of the diesel engine where the swirl chamber system, long recognized as giving a cleaner exhaust in regard to smoke than the direct injection system, is now shown to be superior also in regard to gaseous exhaust emissions, particularly nitrogen oxide. This is illustrated by comparative tests carried out on a single-cylinder engine of 1600cc (96 cu.in.) capacity fitted with a) a Ricardo Comet V swirl chamber combustion system and b) a Direct Injection combustion system. Fig. 1 shows comparative figures for carbon monoxide emissions over the load and speed range of this particular engine. The CO emissions are very low in both cases, in comparison with the figures which would be obtained from a spark-ignition engine, but, even so, those from the swirl chamber system are lower than those from the direct

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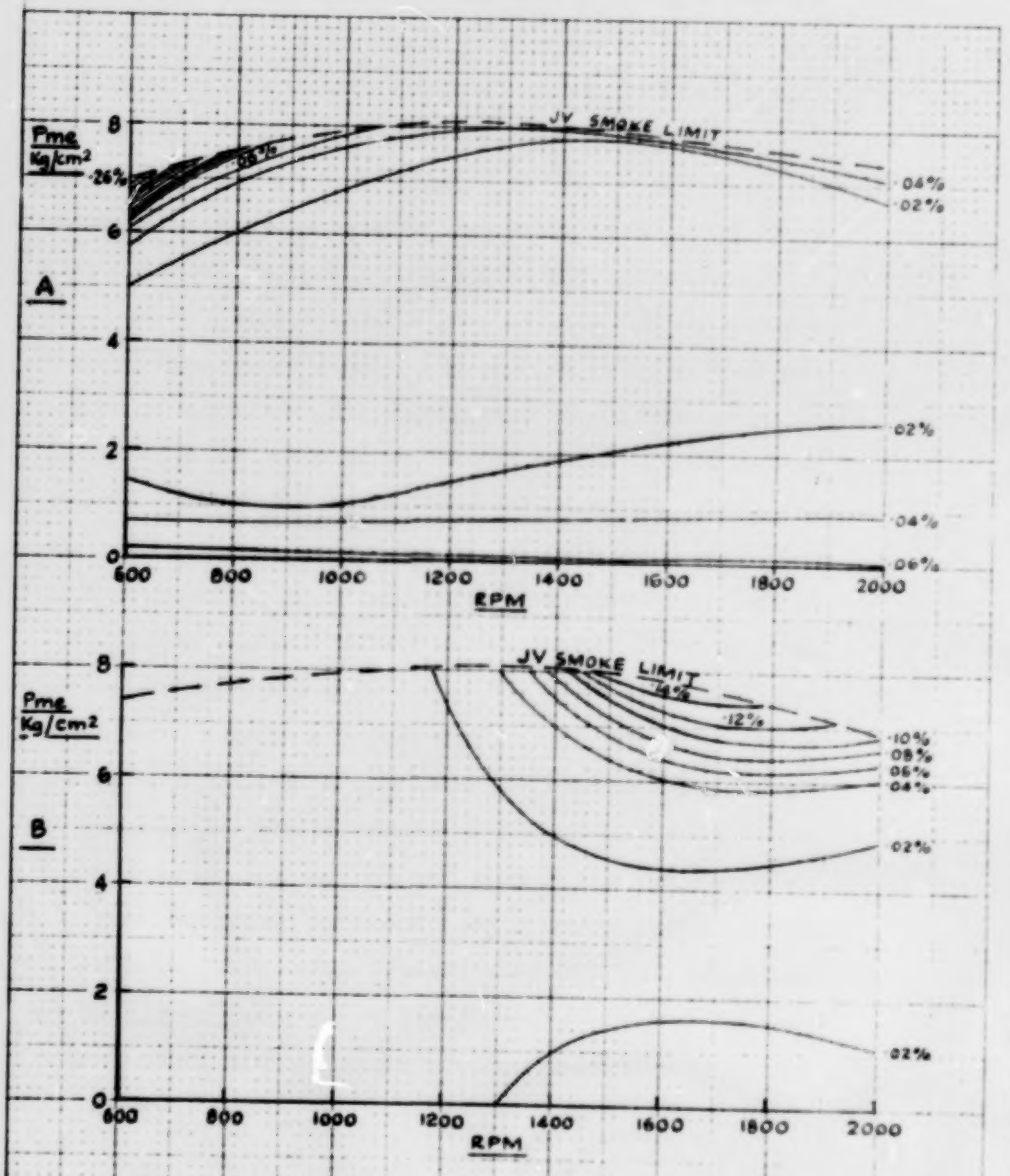
FIG I

EMISSIONS OF CARBON MONOXIDE FROM TWO 120 X 140 mm. DIESEL ENGINES AT OPTIMUM INJECTION TIMINGS.

A. COMET V COMBUSTION CHAMBER
B. DIRECT INJECTION COMBUSTION CHAMBER



injection system. The unburned hydrocarbon figures shown in Fig.2, although they are low in comparison with those from an untreated gasoline engine, give no cause for complacency when we remember the improvements which have been made in the latter unit in recent years. Here again the swirl chamber engine gives lower hydrocarbon emissions than the direct injection unit. Fig. 3 shows the nitrogen oxide figures and here it can be seen that the emissions from the swirl chamber unit are less than half those from the direct injection unit.

An additional advantage of the swirl chamber system is that the injection timing may be retarded from the optimum with a big reduction in NO_x, and incidentally a reduction in noise, but with only a small change in the smoke-limited power, as shown on Fig. 4. A corresponding change in the injection timing of the direct injection engine, although it also reduces the nitrogen oxide concentration considerably, at the same time reduces the smoke-limited power appreciably. There are very good technical reasons for the better performance of the swirl chamber system in regard to nitrogen oxide production. The high rates of fuel/air mixing and therefore of combustion of this system enables retarded injection timings to be used, with a lowering of nitric oxide concentration as noted above, without the severe effect on smoke-limited power experienced with the slower burning Direct Injection unit. Also, with the swirl chamber system, all the fuel is initally injected into only half the air and only afterwards is this rich fuel/air mixture mixed with the rest of the air in the cylinder and burning completed. This results in a lower temperature/volume/time integral for the the swirl chamber in comparison with the direct injection system and this is an important parameter due to the strong temperature

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FIG. 2

EMISSIONS OF UNBURNED HYDROCARBONS FROM TWO 120 X 140 mm. DIESEL ENGINES AT OPTIMUM INJECTION TIMINGS

A COMET V COMBUSTION CHAMBER

B DIRECT INJECTION COMBUSTION CHAMBER

EMISSIONS QUOTED IN ppm. CARBON

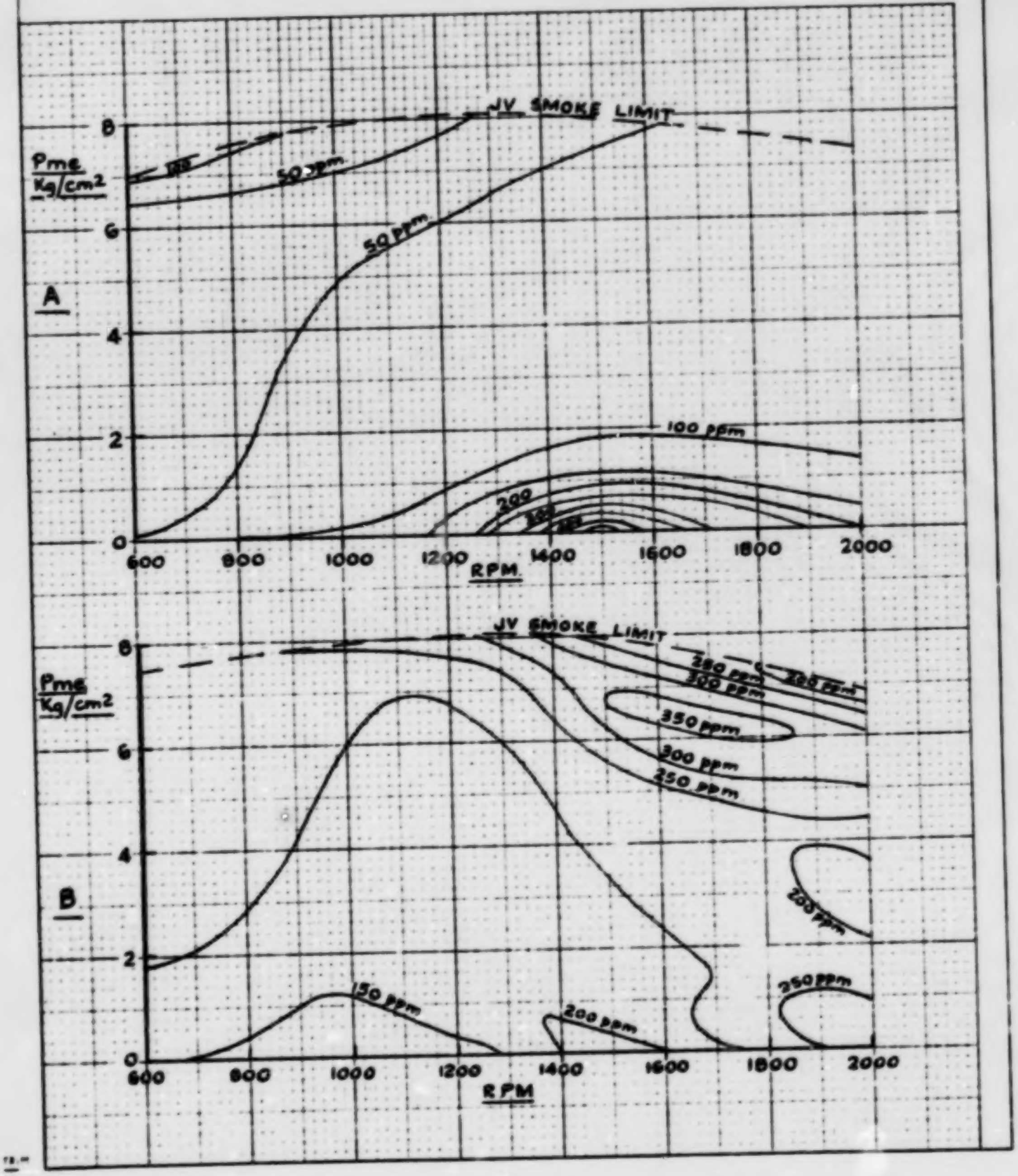

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FIG. 3

EMISSIONS OF OXIDES OF NITROGEN FROM TWO 120 X 140 mm. DIESEL ENGINES AT OPTIMUM INJECTION TIMINGS

A COMET V COMBUSTION CHAMBER
B DIRECT INJECTION COMBUSTION CHAMBER

EMISSIONS QUOTED IN ppm. NO

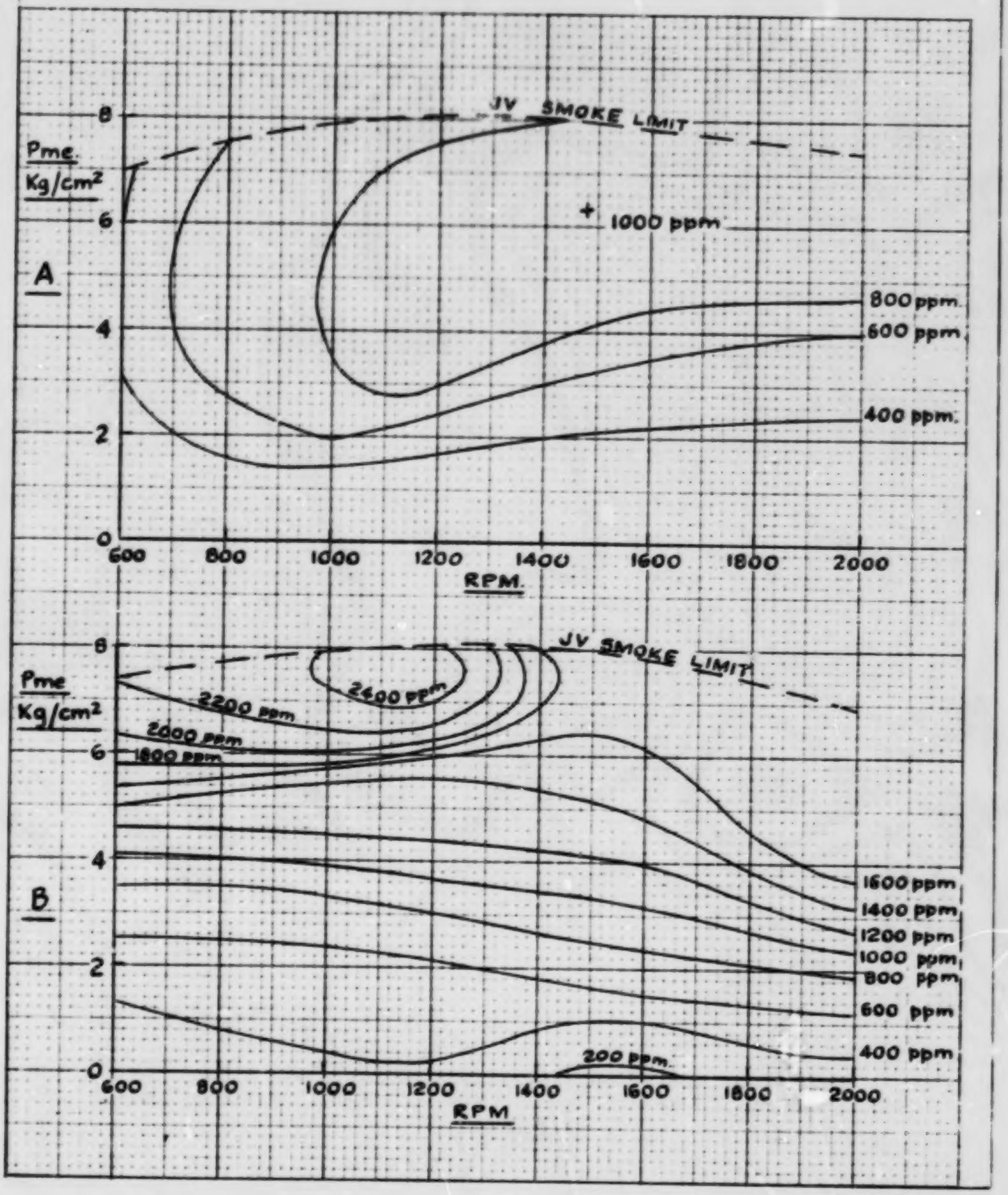

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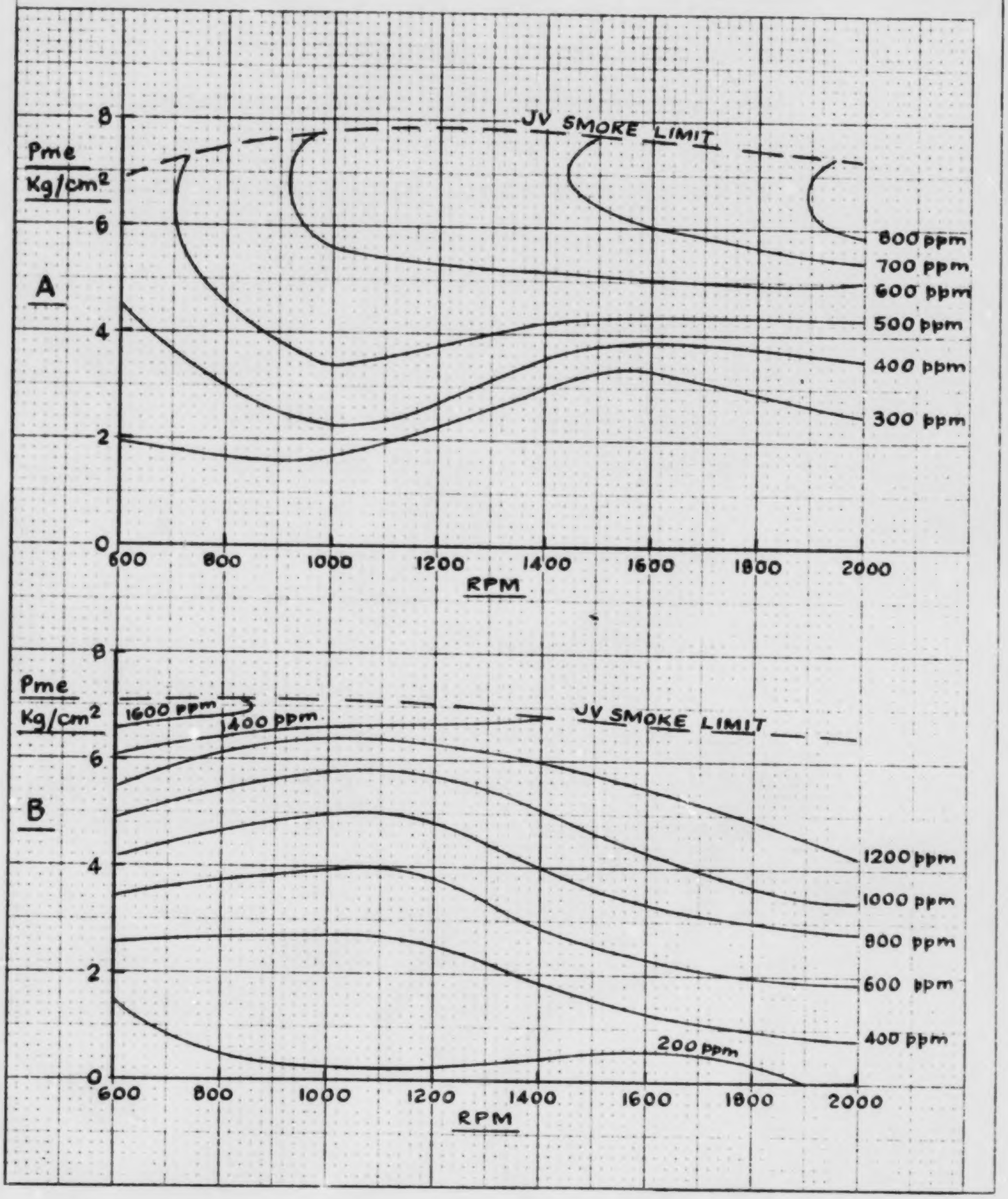
FIG. 4

EMISSIONS OF OXIDES OF NITROGEN FROM TWO 120 X 140 mm. DIESEL ENGINES WITH INJECTION TIMING RETARDED BY 4° FROM OPTIMUM

A COMET V COMBUSTION CHAMBER
B DIRECT INJECTION COMBUSTION CHAMBER

EMISSIONS QUOTED IN ppm NO



dependence of nitric oxide formation.

In the United States, the State of California Air Resources Board has proposed regulations for exhaust emissions from heavy duty diesel engines, as set out in Table 2. The second half of the Table gives results derived from the single-cylinder engine tests, just described. It can be seen that CO is no problem, but that the standard Direct Injection engine is marginal for 1973 and that even the swirl chamber combustion system will not meet the 1975 limits in standard form. It should be mentioned that these single-cylinder engine test results are generally in accord with those obtained from a wide variety of multi-cylinder units.

With retardation of injection timing and possibly with other minor modifications to the engine, it should be possible to get down to 5 gms/hp/h with the swirl-chamber engine, but it is going to be very difficult to do it with the Direct Injection system.

Using EGR with the swirl chamber engine should enable figures of 3 gms/hp/h to be achieved, and somewhat similar or slightly higher values should be obtained with water injection.

Neither EGR nor water injection are very attractive solutions, however, because of their possible effect on engine durability in addition to their influence on performance. A catalytic method of dealing with NO_x, as is proposed for the spark-ignition engine, would appear to be denied us on the diesel, as no one has yet developed a catalytic method of eliminating nitrogen oxide which will work in the oxidising atmosphere almost always present in the diesel engine exhaust. To this extent nitric oxide poses a much

TABLE 2

DIESEL ENGINE EXHAUST EMISSIONS
(gms/hp.h.)

	CO	HC + NO_2
CALIFORNIA PROPOSALS		
1973	40	16
1975	25	5
EUROPEAN DIESEL ENGINES		
1. Direct Injection	2.6	13.4
2. Ricardo Comet Swirl Chamber	3.0	7.3
3. Ricardo Comet 4° Retard	5.5	5.9
4. Ricardo Comet 4° Retard 20% EGR	8.2	3.0
5. Ricardo Comet Optimum Timing 2:1 Water/Fuel Injection	4.0	3.4

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more difficult problem for the diesel engineer than for the gasoline engineer and brings me back to the question I asked right at the beginning. Are we really sure that it is necessary to achieve such low limits of nitric oxide in the engine exhaust?

.. ..

Chapter X

LOW EMISSIONS FROM CONTROLLED COMBUSTION FOR AUTOMOTIVE RANKINE CYCLE ENGINES

by

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ABSTRACT

Rankine cycle engines have a high potential of meeting the emission levels established by the 1970 amendment to the Federal Clean Air Act for the 1975-76 automobile. This paper discusses a Solar research and development program sponsored by EPA/APCO which demonstrates a full scale 2 million BTU per hour working model of a Rankine cycle engine combustor and controls which can surpass the emission goals established.

Special features of the combustor are the unique methods of precisely controlling both the fuel and air to provide optimum flame performance at any engine power level. This paper discusses the special requirements of the Rankine cycle engine and shows why the very wide range of fuel flow required necessitates use of special techniques in fuel atomization, fuel and air control, and aerodynamics. Sufficient discussion is included to show the design methods that are necessary, in general, to achieve low emissions in continuous flow combustion systems. Emphasis is placed on the importance of interfacing a combustion system with other engine parts if a successful low emission, wide turndown ratio combustor working model is to be achieved. Sufficient discussion on combustion kinetics is included to advise on approaches necessary to minimize NO formation in external combustion systems while maintaining high efficiency and low CO and unburned hydrocarbons.

I. INTRODUCTION

The Rankine cycle engine has a high potential of meeting the established 1975-76 emission levels when installed in a family car, thus eliminating an atmosphere of undesirable fumes now commonly contributed to the internal combustion engine. The major portion of such an effort must, however, be devoted to a system to develop a full-scale (2 million BTU per hour) prototype combustor system to demonstrate that the desired low emission levels can be met with the proper operating performance.

Solar has addressed itself to these problems by drawing on its many years of gas turbine experience, studies of combustion kinetics, and applying a novel fuel atomization method. In addition, a precise air-fuel ratio control concept is necessary. The solution lay in continuously modulating the fuel flow to match engine power demands and to similarly control the air-flow to the combustor so that optimum combustion could be obtained under all power level conditions. Such a method of control lay outside the state-of-the-art of present day methods and special techniques had to be applied to the solution of this problem.

Throughout the design, automotive features have been emphasized. Low emission, compactness, high response, low cost potential, and high efficiency have been major considerations in the design selection process. The system described in this report includes all necessary controls to supply and regulate both fuel and air. High response rates are obtained while continuously maintaining an optimum air-fuel ratio with the lowest emissions. An axial blower is used to supply the required air to the combustor. Air-fuel ratio and fuel rates are controlled by a single power demand lever. The effects of air temperature, pressure leakage, flow, speed, and fuel back pressure are automatically compensated by simple mechanical control systems.

II. 1975-76 EMISSION LEVELS FOR MOBILE ENGINES

The goals set for this program were those for a six-passenger automobile vehicle with a maximum test weight of 4600 pounds, tested for emissions in accordance with the procedure outlined in the November 10, 1970 Federal Register (Ref. 10). Values are:

· Hydrocarbons*	1.65	mg/gram fuel	(0.46 gram/mile)
· Carbon Monoxide	16.25	" " "	(4.7 " ")
· Oxides of Nitrogen**	1.38	" " "	(0.4 " ")
· Particulates	0.1	" " "	(0.03 " ")

* Total hydrocarbons (using 1972 measurements procedures) plus total aldehydes. Aldehydes to be 0.16 mg/gram fuel maximum.

** Computed as NO_2.

The task of meeting the goals are graphically displayed in Figure 1, comparing typical emission levels for light duty vehicles of 1960 and 1970 to those published as requirements for 1975-76 in the 1970 amendment to the Federal Clean Air Act.

III. PERFORMANCE GOALS

The basic performance goal of the demonstration system is to develop a full scale (2×10^6 BTU/hr) prototype combustion system which could be capable of achieving the emission levels when such a system is integrated into Rankine cycle engines installed in an automobile. In order to have the widest possible technical significance (to a potential family of yet undefined engines), extreme boundary limits were placed upon the performance goals. These limits appear in the power range, transient response, parasitic power, and volume goals. The importance of transient response was emphasized because previously reported combustor and engine tests with external combustors indicated that on-off cycling and rapid transients were a major source of emissions. The following performance goals were used in synthesizing the combustion system design:

- Heat release: 2×10^6 to 2×10^4 BTU/hr in a maximum of 1.33 cubic feet combustor volume. The unit shall be able to control the time average heat release at any point from 2×10^6 to 2×10^4 BTU/HR with low emissions. This represents a 100 to 1 ratio of heat release between maximum and minimum.
- Fuels: Diesel No. 1, Jet A, or Kerosene
- Rapid transients in firing rates without severe degradation of emission performance. A goal of 50 percent power change per second or 1 to 100 percent power change in 2 seconds has been established to allow interface with fast response flash vapor generators.
- Minimum volume consistent with automotive research goals. Weight factors were considered, but prototype construction practices were used in the early research model.
- Parasitic power losses of less than 2 horsepower without vaporizer.
- Air density variations caused by altitude ranges between zero and 5000 feet, and temperature variations due to variations from 0 to 130° F will be accounted for in the design.

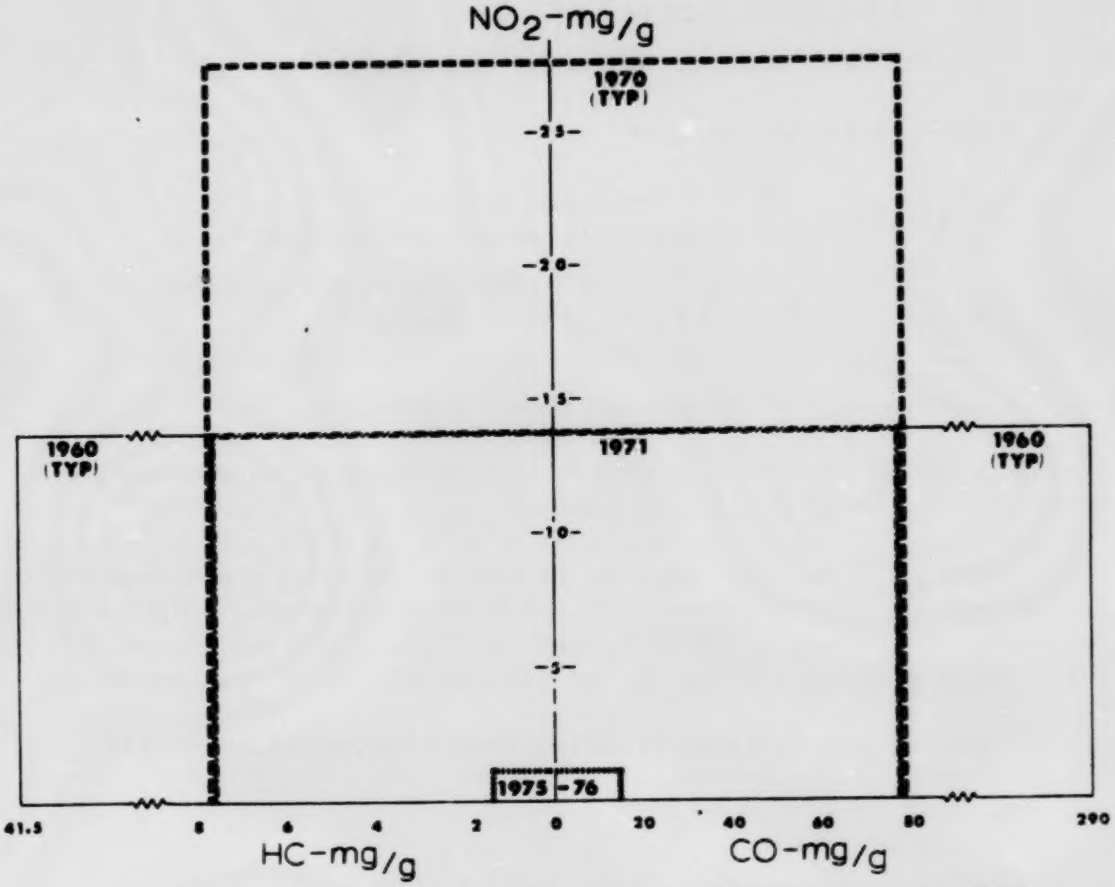


FIGURE 1. COMPARING TYPICAL LIGHT DUTY VEHICLE EMISSION LEVELS TO 1975-76 LEVELS

- Rapid startup to full power in less than three seconds.
- High Reliability and low cost for automotive applications shall be inherent in the design approaches.

IV. COMBUSTION SYSTEM DESIGN

4.1 REACTION KINETICS

Three rules must be obeyed to burn liquid hydrocarbons quickly and efficiently (Ref. 1, 2, 3, and 4):

1) The fuel must be rapidly evaporated

2) The fuel and air must be rapidly mixed.

3) The rate of chemical reaction must be maximized

The rules are satisfied when:

Evaporation	a) Fuel droplet size is small b) Droplet to air relative velocity is high
Mixing	c) Air is injected at high velocity through a large number of holes into a small combustor d) A large number of fuel injection points are used
Reaction	e) The air-fuel ratio is stoichiometric

In practice, unless heat losses are involved, it is necessary to add additional air in order to avoid chemical dissociation losses.

With variations, these rules are the basis for the design of most combustors used in cars, gas turbines, power plants, and home heating systems. Highly efficient combustion can be obtained, free of emissions of smoke, carbon monoxide or fuel. Unfortunately, this design method can result in high emissions of nitric oxide (NO).

The rate of formation of NO has been found accurately represented by

$$\frac{dNO}{dt} = 6.62 \times 10^{14}\, e^{-67,000/T} \left[\frac{(N_2)(O_2)^{1/2} P^{1/2}}{T^{1/2}} \right] \qquad \text{(Ref. 5)}$$

where NO, N_2, and O_2 are concentrations of nitrogen, oxygen, and nitric oxide in mole fractions

t is time in seconds

T is temperature, °K

P is pressure in atmospheres.

Time can be assumed infinite after one second and the resultant equilibrium values of nitric oxide are shown in Figure 2. Peak NO emissions occur at slightly greater than stoichiometric air and fall rapidly when excess fuel is present. The greater the time spent at top temperature at any air-fuel, the greater is the amount of NO formed as shown in Figure 3.

In a reducing atomsphere, the lack of excess oxygen permits the use of catalysts to break down any nitric oxide that is formed and so in an automobile, presents additional flexibility in control (Ref. 6). In combustion systems using excess oxygen, catalysts are not practical and the only method of control is to limit the initial NO formation. The rule governing such a combustor design is:

1) Reaction time must be small in those parts of the combustor where air fuel is near stoichiometric.

   Therefore, combining the rules for efficient and fast combustion with minimum NO formation, we would have a combustor of small size, having a high pressure loss, a multiplicity of fuel injection points and an air-fuel distribution as follows:

   a) A well mixed primary fuel-air zone having excess fuel, of sufficient volume to permit maximum reaction of fuel
   b) A secondary fuel-air zone where excess air is rapidly and uniformly added and having no more volume than is necessary to assure completion of the combustion reaction
   c) A tertiary fuel-air zone (when needed) where any required excess air is added and where no reaction occurs.

Such rules would apply in gas turbines, Rankine cycle engines, and any other combustion system where air-fuels much greater than stoichiometric are involved. In Rankine cycle external combustors operating near stoichiometric, some additional control is permitted because heat losses to the boiler walls can bring down flame temperatures substantially. Often, in such applications, inlet air preheat is used which has the reverse effect. In gas turbines, because wide operating conditions result in large changes of air-fuel, there is only one unique point (usually a high power condition) where the rules can be ideally maintained. A penalty in increased emissions at other operating conditions is ordinarily accepted and this is the cause of high hydrocarbon emissions near airports because of ground taxi or start up. Power plants in autos do not have any unique operating point and typically have to start frequently and undergo rapid power level changes. In a gasoline engine, it is therefore necessary to precisely control the air-fuel ratio by means of a carburetor. It is also considered essential in Rankine engines used in an auto that the airflow into the combustor be controlled so that the rules for minimum emission are held at any fuel flow.

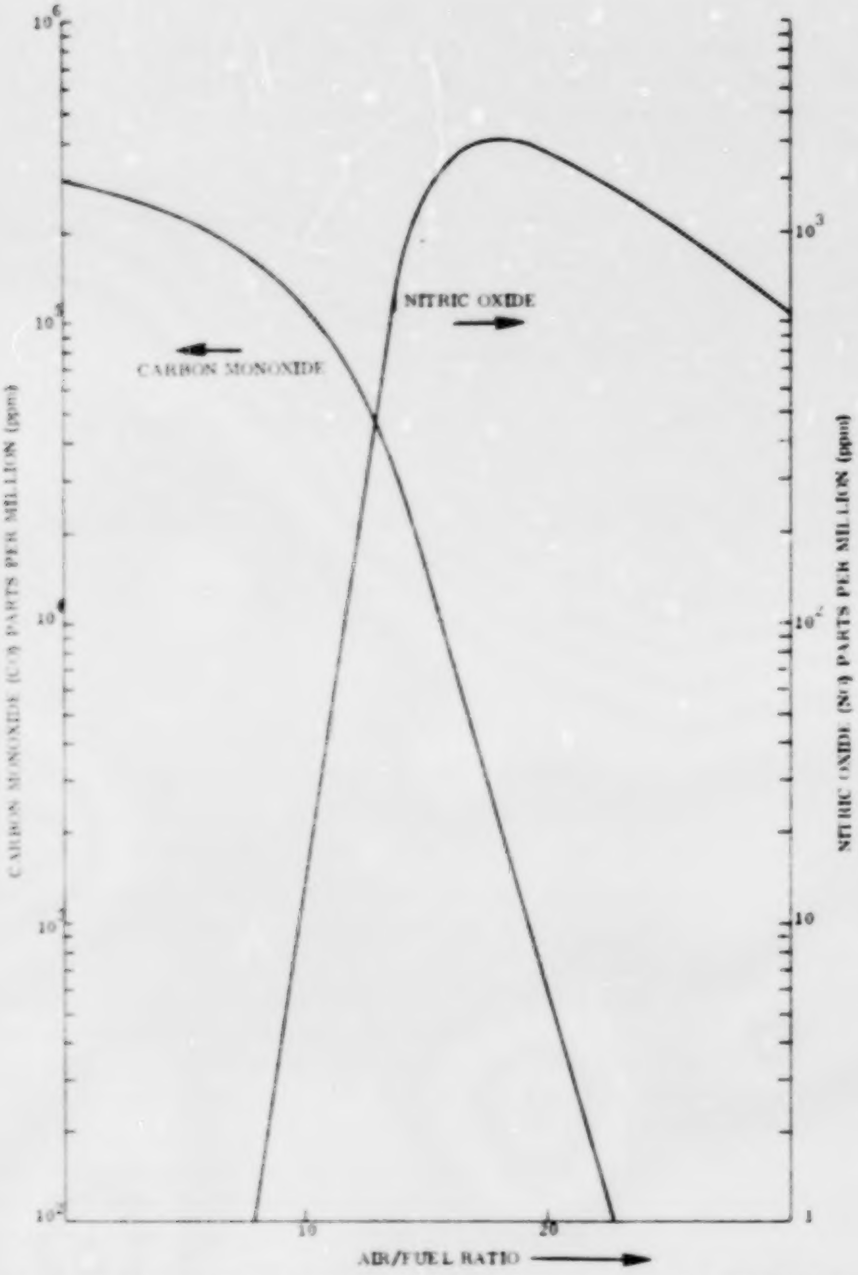


FIGURE 2. EQUILIBRIUM CONCENTRATIONS BY VOLUME OF CARBON MONOXIDE AND NITRIC OXIDE AS A FUNCTION OF AIR/FUEL RATIO

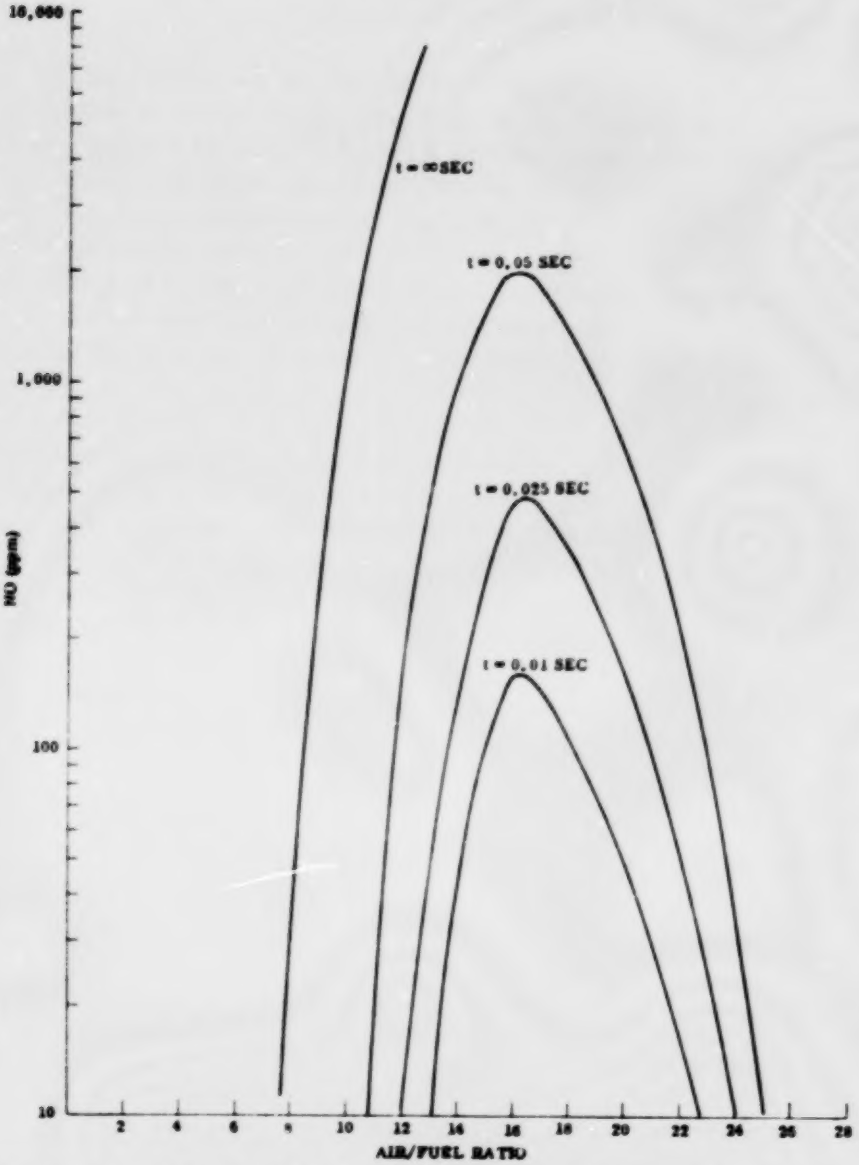


FIGURE 3. EMISSION OF NITRIC OXIDE AS A FUNCTION OF AIR FUEL RATIO AND TIME

4.2 AIR AND FUEL CONTROL SYSTEM DESIGN APPROACH

The control system was designed to regulate the flow of fuel and air to set power outputs as a function of an input command from vaporizer control. A fuel ratio range of 100 to 1 has been established as a design goal for this demonstration system. Since present state-of-the-art controls have difficulties providing turndown ratios of more than 15 to 1, this particular requirement has been recognized as a critical design area requiring unique solutions. In order to optimize the low emission characteristics of the combustor, it is essential that throughout the entire power range (1 to 100%), an optimum air-fuel ratio be maintained. An optimum envelope in which the air-fuel ratio should be centered across the entire power range is discussed in detail in Section 5 and forms the basis for the control.

By directly mechanically coupling a fuel metering valve and the large diameter air metering plate (Fig. 4), the fuel flow area and the air flow area can be kept in an exact ratio correspondence. Once the area ratios have been fixed, it is only necessary to regulate pressure drops in order to maintain the desired weight flow ratios. Figure 5 shows the overall mechanical arrangement of the demonstration combustor system.

For low pressure rise, air compression can be neglected and the weight flow relationships can be written as:

$$W_a = K_a A_M \sqrt{\frac{P_1}{T_1}(P_1 - P_2)} \qquad (1)$$

$$W_f = K_f A_f \sqrt{\Delta P_f} \qquad (2)$$

where W_a = mass air flow into combustor

A_M = air metering valve area

P_1 = upstream pressure (blower discharge)

P_2 = downstream pressure

T_1 = air temperature at metering valve

W_f = fuel mass flow

ΔP_f = pressure drop across fuel valve.

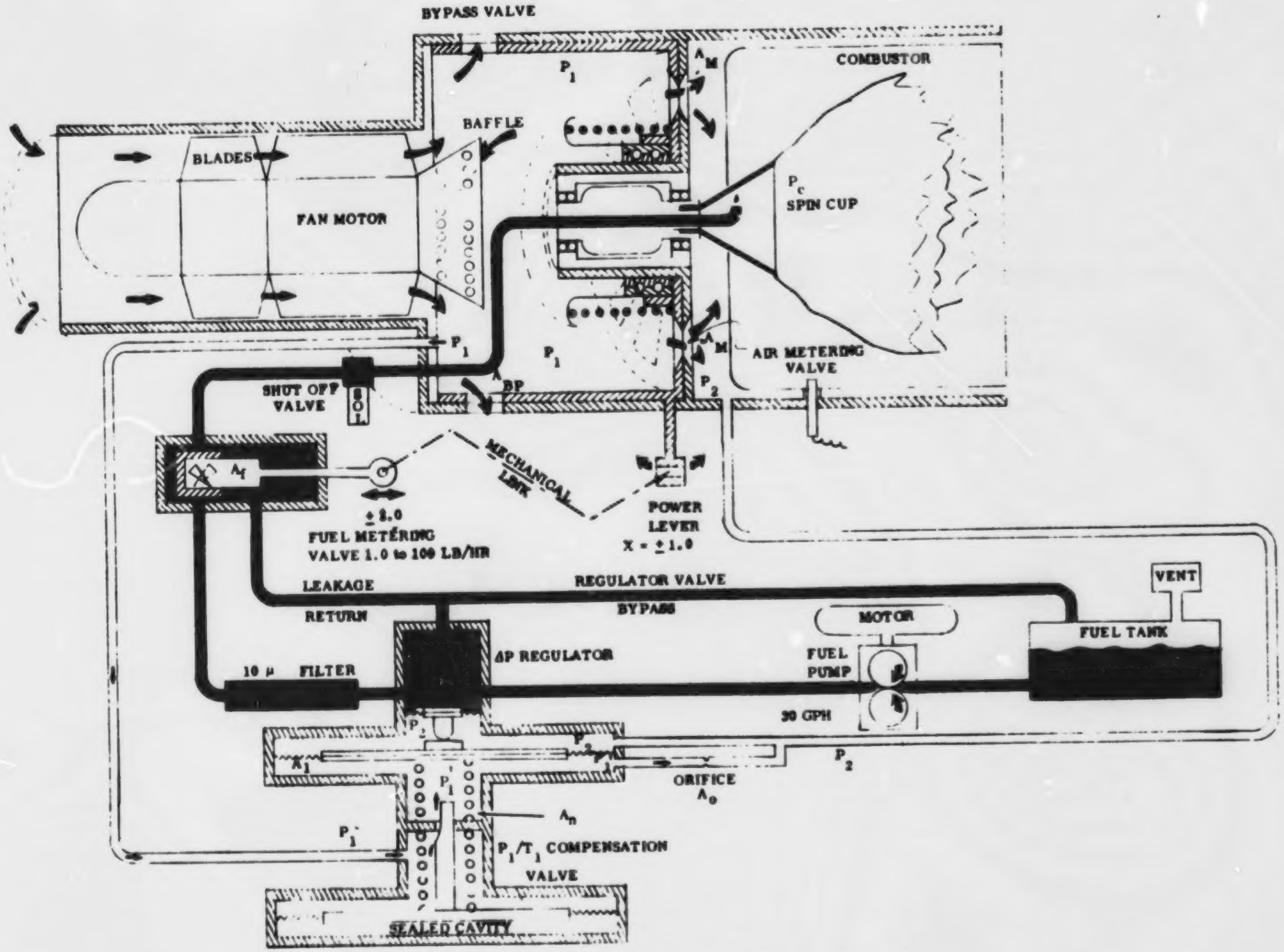


FIGURE 4. CONTROL SYSTEM SCHEMATIC

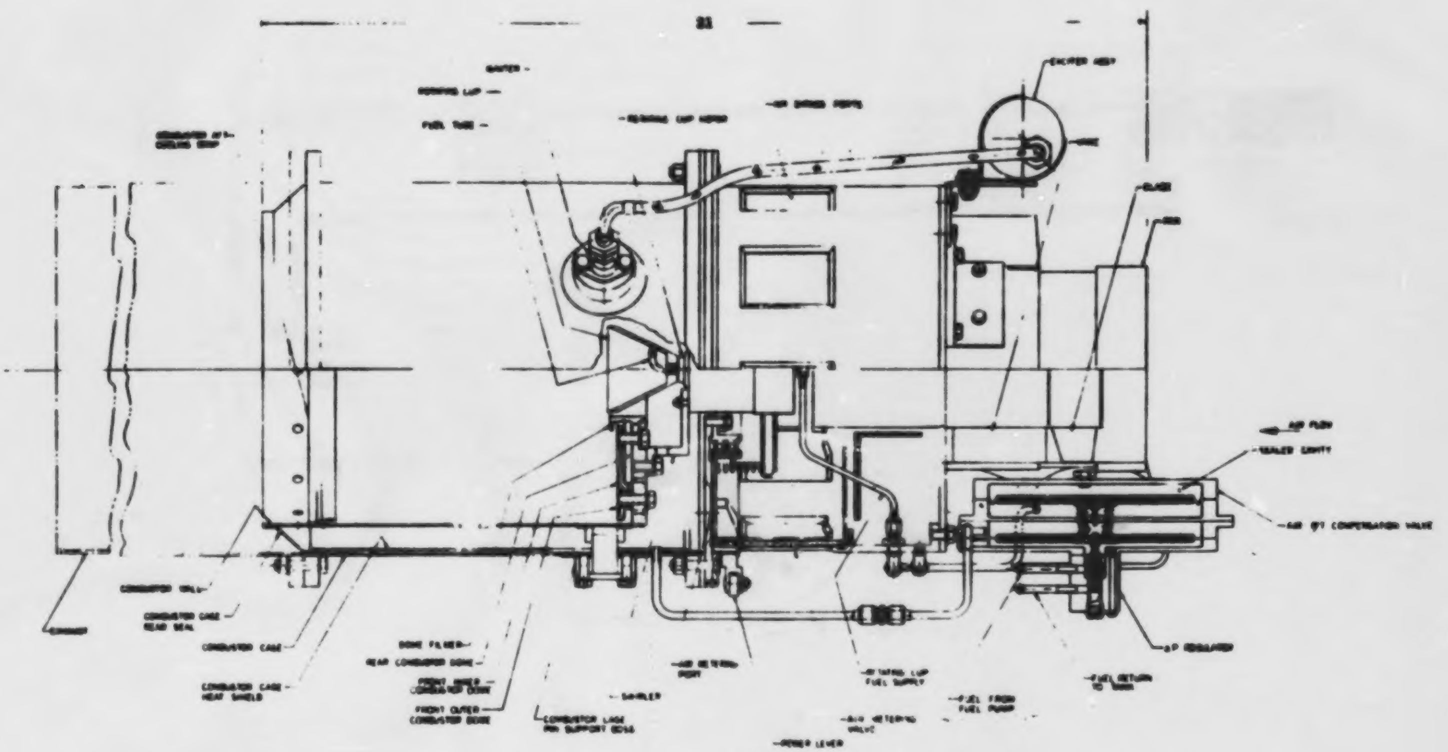


FIGURE 5. DEMONSTRATION SYSTEM

If the inlet pressure (altitude) and temperature are constant, a simple relationship can be written for the air fuel ratio:

$$\frac{W_a}{W_f} = K\sqrt{\frac{P_1 - P_2}{\Delta P_f}} \qquad (3)$$

To ensure the air-fuel ratio remains a function of flow area, only the pressure drop across the fuel valve will be controlled proportional to the pressure drop across the air metering ports. A ΔP_f regulator valve performs this function by a force balance across diaphragms. Operation of this component is illustrated in Figure 4. P_1 pressure is connected (through a density compensator) to the bottom side of diaphragm A_1. Downstream pressure (P_2) is connected to the opposite side of the diaphragm. Fuel system pressure is regulated by a flapper valve that recirculates excess fuel back to the tank. A small diaphragm on the fuel side balances the pressure across the air metering ports A_M against the fuel pressure. If the air pressure drop increases (causing W_a to increase), the force across the system becomes unbalanced and the fuel pressure regulator moves up; reducing the by-pass flow, and thus increasing the fuel pressure and its mass flow across the fuel metering valve. Since the force balance must be maintained across the ΔP_f regulator, we have:

$$\sum \text{Forces} = A_2\,(\Delta P_f - P_2) = A_1\,(P_1' - P_2)$$

and where ΔP_f is large compared to P_2

where $P_1' = P_1$ (times) ambient air temperature and pressure compensation factor

we have $$\Delta P_f = \frac{A_1}{A_2}\,(P_1' - P_2) \qquad (4)$$

Thus equation (3) can be rewritten as

$$\frac{W_a}{W_f} = K\sqrt{\frac{(P_1 - P_2)\,A_2}{(P_1' - P_2)\,A_1}} = \text{constant}$$

Blower speed changes due to voltage or load variations, blower efficiency reductions due to fouling or wear, and by-pass valve leakage variations are automatically compensated by maintaining the ΔP_f as a function of the air valves pressure drop.

It can be seen from equation (1) that the air flow is a function of

$$\sqrt{\frac{P_1}{T_1}}.$$

Thus, an altitude and ambient air temperature correction is applied by adding a P_1/T_1 compensator (Fig. 4). Blower discharge pressure (P_1) is admitted to the ΔP regulator by a contoured needle valve. This valve is positioned by a force balance across a diaphragm having P_1, T_1 on one side and cavity sealed with air at standard conditions. Thus, as the air temperature increases, the sealed air will expand and move the valve upwards. This action will increase the pressure drop across the contour valve and thus reduce P_1 to a corrected value of P_1' on the bottom side diaphragm A. As P_1' decreases, the fuel pressure will drop, producing the desired fuel flow reduction as the ambient air temperature increases.

Air to the combustor is metered across a series of twelve ports cut into two plates (Fig. 6). An input lever rotates one plate with respect to the stationary backup plate. The metering port areas are caused to open or close as the input power lever rotates matched ports in each plate. By maintaining a known relationship to power lever position and metering port area, the port contour can be arranged to provide near linear control as a function of power lever position and voltage with the ΔP regulator compensating for differences in the range of 1 to 2.5 inches of $H_2O\Delta P$.

The capability of the fuel regulator to control fuel pressure levels proportional to differential metering valve pressures of 1 to 2.5 inches of H_2O is vital to obtain a wide flow range control with high response that maintains an accurate air fuel ratio from 1 to 100 percent power demands. Initial analysis indicated that the use of motor voltage (speed) regulation could not meet the range accurately or the stringent transients improved by the Federal driving standards. Inertial lags of the motor would keep it out of synchronization with the fuel flow for an unacceptable percentage of the standard driving cycle. By maintaining the fuel pressure proportional to the ΔP across the air metering plates, the demonstration system allows limited voltage control (to reduce parasitic losses and noise at normal driving speeds) with precise air-fuel ratio control under transient conditions. If the by-pass valve is maintained closed for the power range from 100 to 70 percent, the air flow into the combustor can be accurately controlled by proportionally lowering the voltage to the blower motor while the regulator compensates for the reduced pressure differential across the air metering valve at lower speeds. Since a mechanical linkage maintains a known ratio of air valve area to fuel valve area, the regulator will automatically maintain the air fuel ratio as the power is reduced. A limitation on this sytem is the lower differential pressure signal that can maintain accurate fuel pressure regulation. Analysis and test has indicated that 1.0-inch of H_2O is a reasonable lower limit. Using this value, the part load

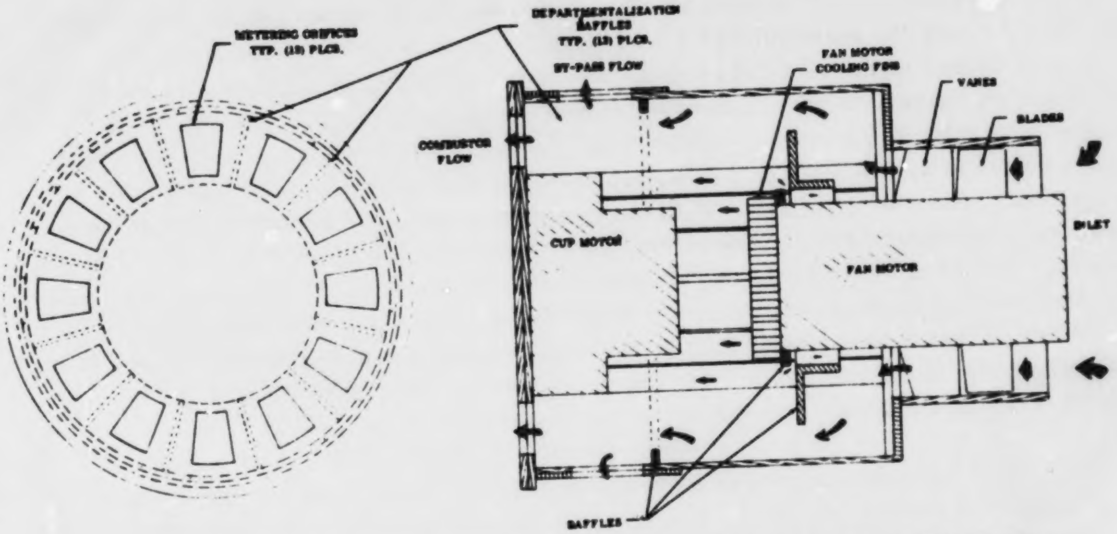


FIGURE 6. SWIRL ELIMINATION AND PRESSURE EQUALIZATION BAFFLE ARRANGEMENT

power demands of the combustor system can be established. Although the combustor requirements are moderate (1.25 HP for the optimum configuration and 2.3 HP for the demonstration system), addition of a typical boiler configuration can make parasitic power losses at part loads unacceptable. If a boiler of the type described in Reference (11) was used, an additional 2.4 horsepower would be required. Total parasitic power levels would then be as high as 3.65 horsepower. If the system were to require this high power level from full power down to idle condition, a highly undesirable condition would exist. Lowering the input voltage across a voltage range that allows accurate ΔP compensation by the regulator will eliminate the motor inertial speed lags since the air fuel ratio changes are continually maintained by the regulator. Power reduction by operating at 1.0 inch of H_2O can be estimated by simple calculations based on fan laws for a series wound motor. For voltage changes of approximately 2 to 1, the following relationships give reasonably accurate results.

For a given voltage change V_1 to V_2

flow Q: $Q_1/Q_2 = V_1/V_2$

pressure p: $P_1/P_2 = (V_1/V_2)^2$

fan BHP is: $BHP_1/BHP_2 = (V_1/V_2)^3$

By maintaining the bypass valve closed and reducing the voltage to the motor, the flow through the metering valve will drop as a function of both the motor voltage and area change of the metering valve. Pressure will drop as a function of the square of flow. By using 1.0-inch as the lower limit of adequate control, the bypass valve can be maintained in a closed position until the voltage is reduced to 1/2 or $0.707V_1$. Since power is approximately a cubic function, we will obtain $(0.707)^3$ (BHP), or a 65 percent reduction in power. With the optimum system and a relatively high air side boiler pressure drop, a parasitic power loss reduction of approximately 2.4 horsepower. Total power with this high pressure drop boiler would be approximately 1.25 horsepower at vehicle power demands below 35 percent. One difficulty with this approach is that it is more difficult to make the system linear. However, since this is not a driver input command linearity, it is of secondary importance if the correct air-fuel ratios can be maintained.

Fuel must be accurately metered from 109 to 1.0 pounds per hour. At low fuel rates (1.0 pounds per hour), the flow is approximately two drops per second. Standard valves do not have sufficient linear range to accommodate these severe requirements. Additionally, the valve should not be sensitive to temperature induced fuel viscosity changes. A new approach to this problem has been taken by the application of a dual slotted shear valve (Fig. 7) consisting of two flat (ground and lapped) plates with matched contour slots 90 degrees to each other. At the intersection of the two slots, a square orifice is formed whose area is a function of the relative position of the top movable plate. The square shape (and thus the discharge coefficient) can be maintained constant throughout the entire 100 to 1 area ratio. Since the plates are in contact, fuel will flow only through the slot in each plate and not between the plates, thereby reducing the clearance leakage path to the microfinish of the contacting surfaces. Figure 8 shows the assembled fuel valve of the demonstration system. A drain groove is provided between the upstream pressure and the metered outlet fuel passage, thereby reducing the potential leakage pressure to the level of the frictional flow drop to the spin cup. Since this is normally less than 0.5 psid, resulting leakage of metered fuel into the drain system will probably be negligible.

The size of the orifice slots used is a trade-off between four factors.

- Fabrication capabilities - requires large dimensions
- Contamination - requires large dimensions
- Backpressure sensitivity - requires high pressure and thus, small sizes
- Temperature sensitivity - it is desired that changes in fuel temperature have little effect of the coefficient of discharge. This factor requires a high Reynolds number and thereby small orifice dimensions.

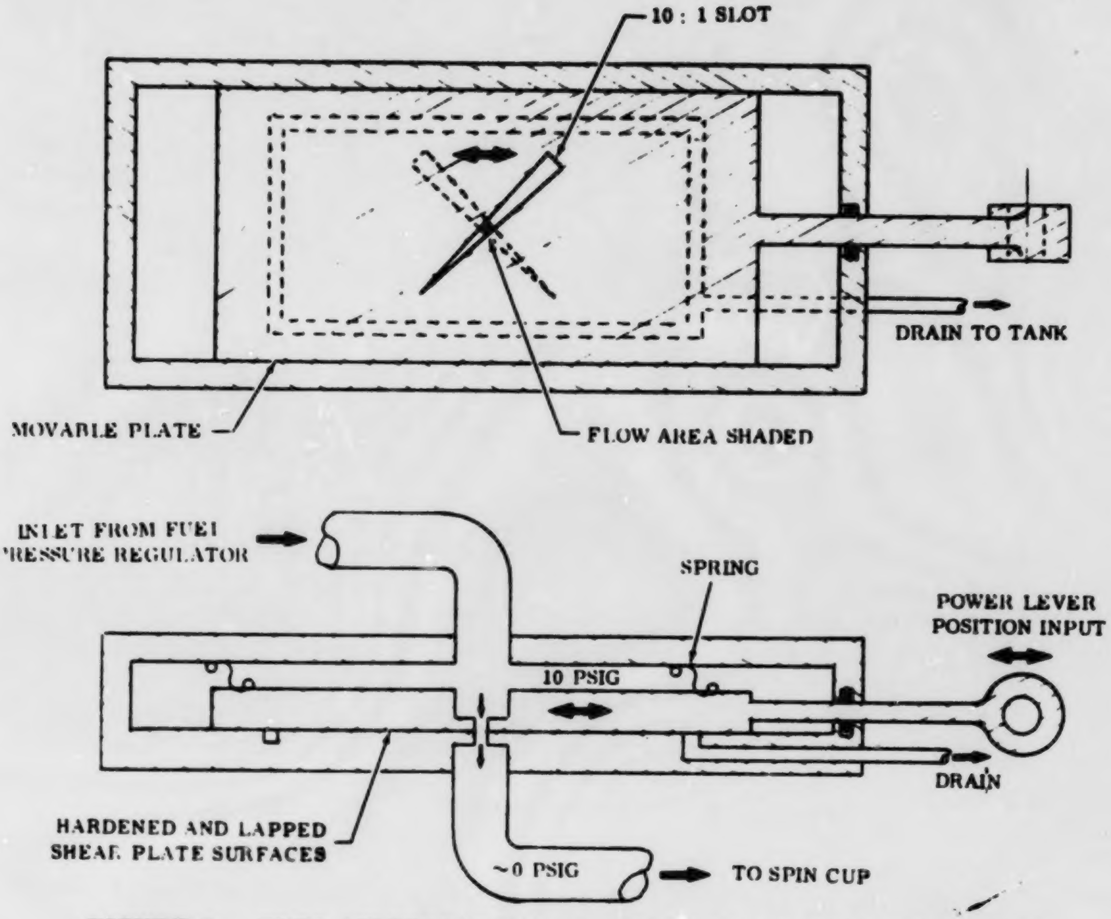

10 : 1 SLOT
DRAIN TO TANK
MOVABLE PLATE
FLOW AREA SHADED
INLET FROM FUEL PRESSURE REGULATOR
SPRING
POWER LEVER POSITION INPUT
10 PSIG
DRAIN
HARDENED AND LAPPED SHEAR PLATE SURFACES
~0 PSIG
TO SPIN CUP

FIGURE 7. FUEL METERING VALVE CONCEPT FOR 100:1 TURNDOWN

4.3 FUEL INJECTION, ATOMIZATION AND IGNITION

The requirements of the fuel injector are:

1) Small droplet size
2) Reliable and low cost
3) Multiple injection points
4) Precise spray angle.

In detail, these requirements entail:

1) Small droplet size must be obtained with kerosene at any fuel flow between 1.0 and 109 pounds per hour with fuel viscosity, varying with ambient temperature, ranging from less than 1 to over 16 centistokes.

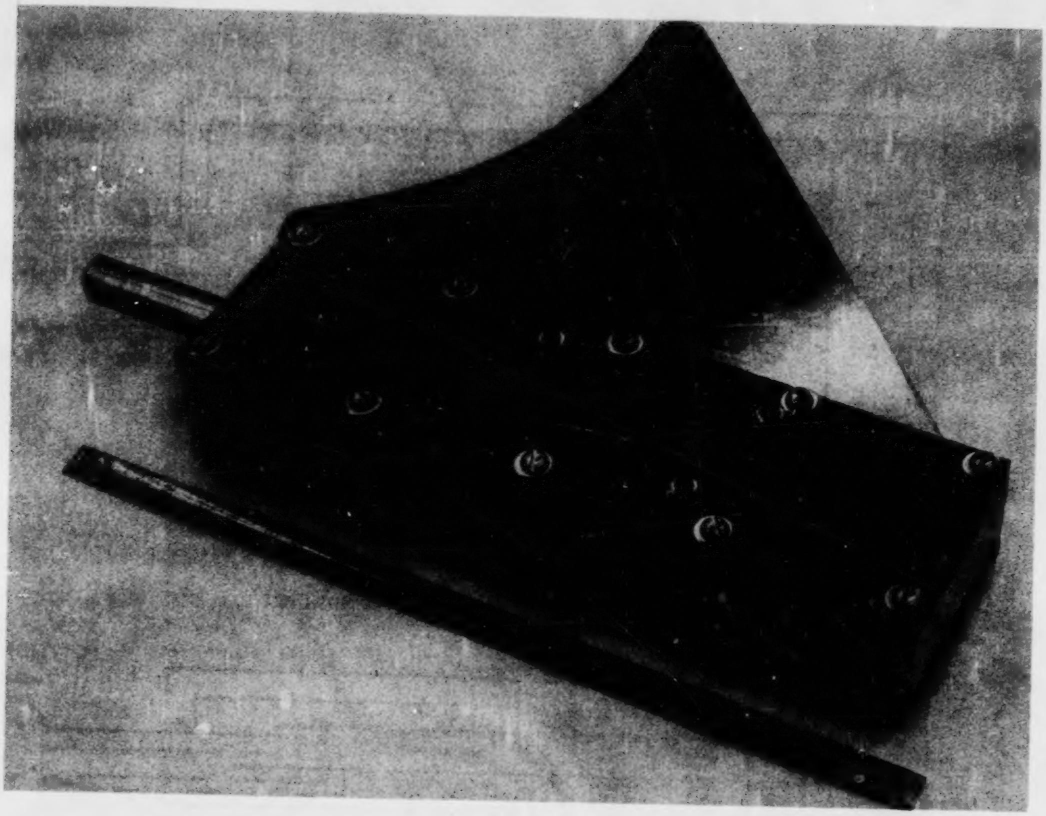


FIGURE 8. FUEL METERING VALVE ASSEMBLY

2) The injector must be of proven design, unsophisticated, not requiring a high fuel pressure pump or other complex or costly auxiliaries. Fuel orifices must be large to prevent contamination.

3) The object of multiple injection is to provide the maximum interface of fuel and air and hence provide fastest and most uniform mixing so important for optimum air-fuel ratio and fastest reaction rate. Packaging in an auto requires, at this stage, maximum design flexibility. If the combustor were mounted horizontally, being at least 10 inches in diameter, the manifold head effect would require, for good fuel distribution, a fuel pressure of over 10,000 psi, and is impractical. The mixing must therefore be mainly achieved by good design of air injection and providing the maximum surface of fuel spray to the air.

4) Precise spray angle is necessary to control fuel air mixing and also to obtain ignition by assuring that the spark is adjacent to but not smothered by the fuel droplets. Ignition is easiest at low velocities and hence should be done at 1.0 pound per hour. (An ignition failure at 109 pounds per hour would be a dangerous fire risk and serious polluter.)

There are four principal classes of fuel injectors (1) fuel pressure, (2) air assisted, (3) rotary, and (4) vaporizer (Ref.). The first three operate on the same principle; i.e., the fuel is presented to the air at a high relative velocity which shatters the fuel into small droplets. Vaporizers rely on heat input, usually from the flame, to vaporize the fuel. A design study, adumbrated below, indicates the rotary injector to be the best choice.

1) Pressure injectors cannot atomize fuel flows as low as 1 pound per hour except when nonviscous fuels are used. Orifice size would be minute and fuel pressures above 250,000 psi are needed. Spray angle is extremely sensitive to viscosity.

2) Air assist injectors use high velocity air to atomize the fuel. There are two types, one using high air volume and low pressure (8-inch water) (Ref. 2). The other using high air pressure (several psi) and low volume. Some of these injectors use such high air pressures that sonic velocities are reached (Ref. 7). The high air volume system is not practical, because at low fuel flows, the air flow required would be far too much for combustion. The high pressure system injects air at such high velocities (400 ft/sec) that at low fuel flows, where combustor air velocities are of the order of 1 ft/sec, a flame could not be stabilized. In addition, the spray angle is sensitive to viscosity, especially at low fuel flows and, coupled with the high air velocities, makes reliable ignition doubtful. Unless high fuel pressure is used at high fuel flows (with resultant small orifice sizes) air flow has to be high and substantial power is required. Both types of injectors need power absorbing and costly auxiliaries and involve the use of small air and fuel passages.

3) A rotary atomizer does not need auxiliaries as it can be mounted directly onto the fan required to deliver combustion air. No fuel pressure or fuel orifices are needed as the fuel is passed through a large tube (3/16-in. diameter) that passes through the center of the fan motor shaft, and trickled onto the cup surfaces. The fuel adheres to the cup wall and is ejected from the outer lip of the cup at a velocity of about 40 feet/second. The power required to drive the cup is negligible. Fuel spray angle is extremely precise and independent of fuel flow or viscosity (no ignition failures have occurred in tests to date).

4) A vaporizer requires an auxiliary source of heat for ignition and initial flame propagation. The response is slow and the heat losses involved at 1 pound per hour would prevent combustion unless auxiliary heating were continuously supplied. The large variation in heat release makes it

most difficult to prevent overtemperature of the vaporizer at high flow rates while maintaining sufficient heat at low flows. It is sensitive to fuel volatility, hence requires a more precisely refined fuel type than the rotary injector and flame performance will be significantly influenced by ambient day temperature (in a cold start it will smoke). These objections apart, a vaporizer is a desirable system. This is because the absence of fuel droplets avoids evaporation delay and allows a more uniform mixing of fuel and air.

4.4 COMBUSTOR DESIGN

The combustor must be small in size in order to fit in an auto and also to limit combustion time. The use of high combustion velocities requires high air injection velocities for proper mixing and hence high pressure loss. A limit of auxiliary fan power prevented a pressure loss of more than 8 inches of water being available. This dictated the use of a relatively large combustor (Fig. 5 and 9). The combustor was a scaled up version of a similar rotating cup combustor used in a Solar 10 KW gas turbine under development. Construction techniques are typically as used in gas turbines, except that the best available oxidation resistant material was used (Hastelloy X). This permitted operation of highest possible wall temperatures and minimized film cooling. Film cooling, by its very function, implies low rates of mixing (Ref. 9) and breaks a fundamental rule in mixing of fuel and air in that it must be good.

Air was arranged to enter the combustor through various ports, the air-fuel ratio calculated, and computer runs made of the resultant emissions. Figures 10 and 11 are typical results. Combustion time at full fuel flow is nominally 0.01 second, and this increases to 1.0 second at minimum fuel flow. The resultant increase in NO emissions at low flows is clearly seen in Figure 11.

V. DEVELOPMENT TESTING

5.1 COMBUSTOR TESTING

Initially, the fan was not used and air was supplied by a remotely located air compressor that supplied air via a plenum to the combustor. This guaranteed uniform air distribution. For convenience of test, the cup was driven by a small electric motor independent of the fan (Fig. 12).

After development of the combustor and cup, it was possible to maintain an efficient flame over a range of fuel flows from over 109 pounds per hour to less than 0.3 pounds per hour, provided that the air-fuel ratio was kept within reasonable limits (Fig. 13 and 14). A range of air-fuels were tested and Figures 15 and 16 show the emissions of CO and NO at the optimum air-fuel together with the results of a 10

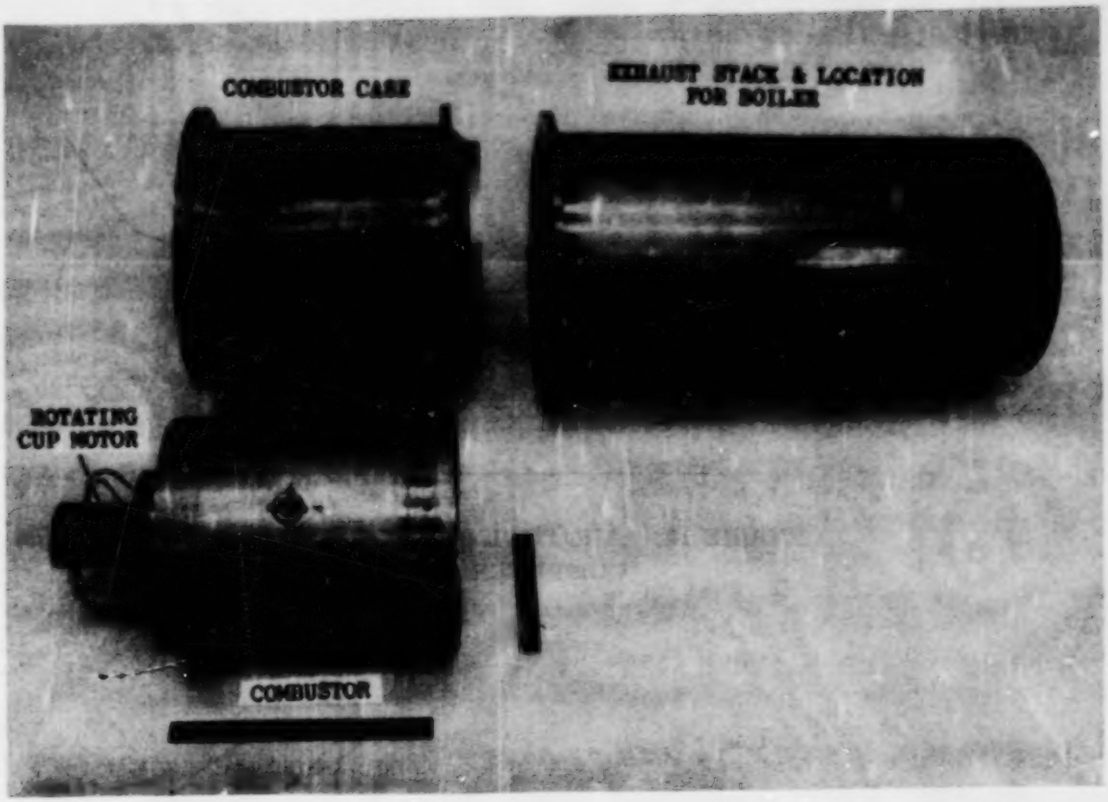


FIGURE 9. SIDE VIEW OF ROTATING CUP COMBUSTOR ASSEMBLY

percent error in air-fuel. Figure 17 shows the optimum air-fuel required for minimum emissions. During the testing, hydrocarbon emissions were monitored, at optimum air-fuels, below background levels. This is not surprising in view of the close proximity of the test facility to the San Diego airport (200 yards). The results are noteworthy in that:

1) The NO emissions fall off sharply at low fuel flow and the CO emissions increase. This is due to the high heat losses from the flame at these conditions.

2) The large increase in emissions in the mid-fuel flow range when any deviation from optimum air-fuel occurs. This was found caused by poor fuel atomization that occurred in this region and was subsequently eliminated by design changes.

3) The large variations in optimum air-fuel as a function of fuel flow. At high fuel flows, this is partly attributable to the poor fuel atomization which varied as a function of fuel flow and due to the changes in fuel-air

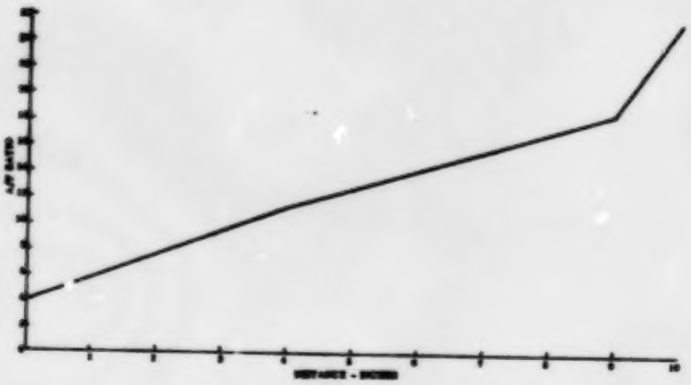


FIGURE 10. AIR/FUEL RATIO ALONG COMBUSTOR FROM COMPUTER ANALYSIS

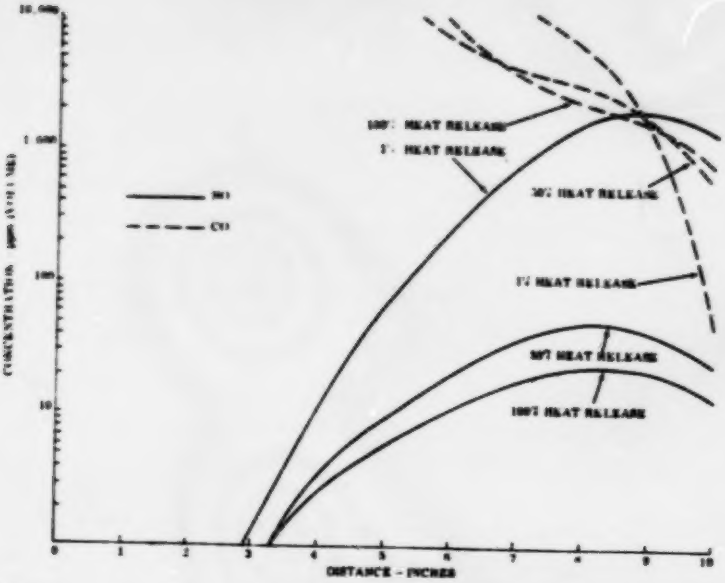


FIGURE 11. NO$_x$, CO CONTENT FROM COMPUTER ANALYSIS

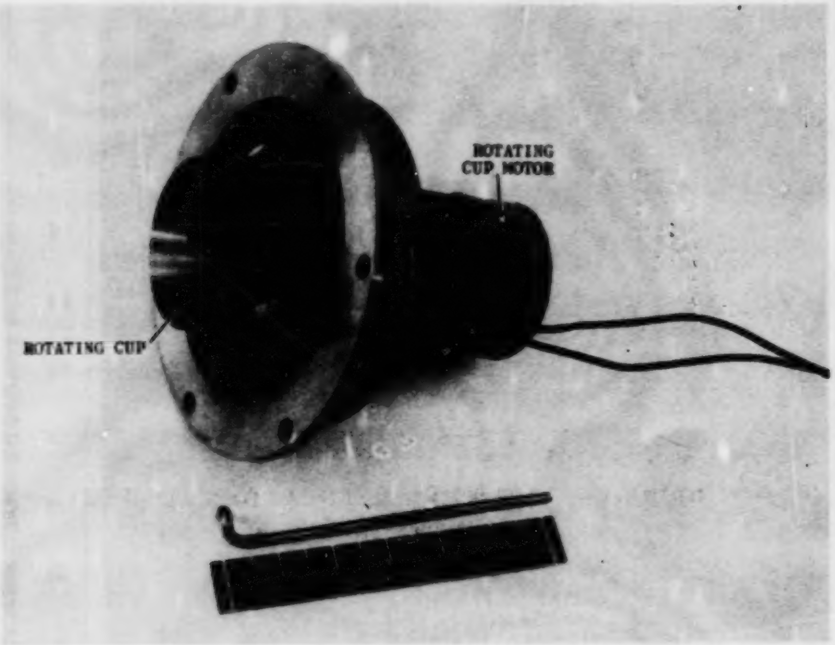


FIGURE 12. ROTATING CUP AND MOTOR ASSEMBLY

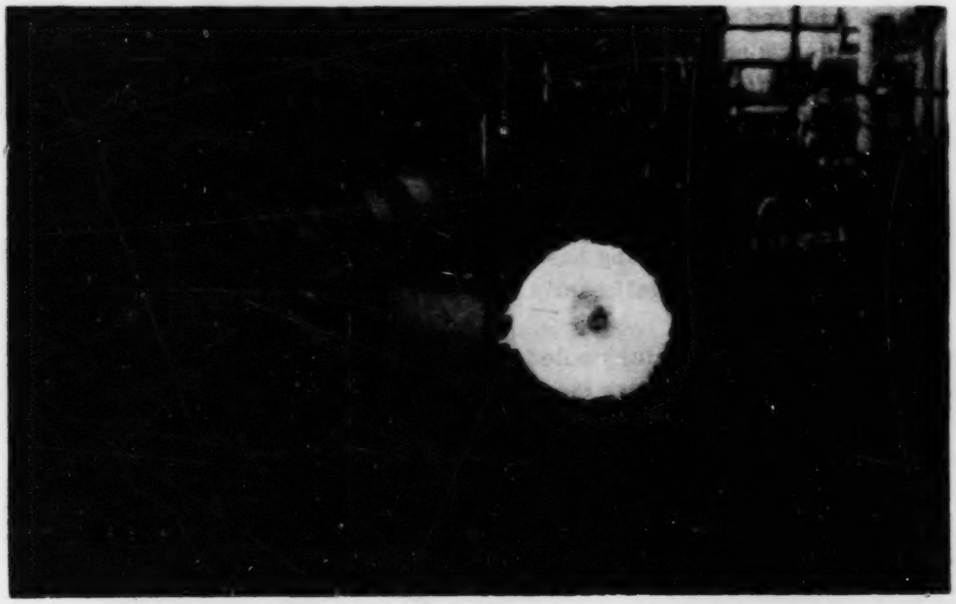


FIGURE 13. FLAME AT 109 POUNDS/HOUR FUEL FLOW

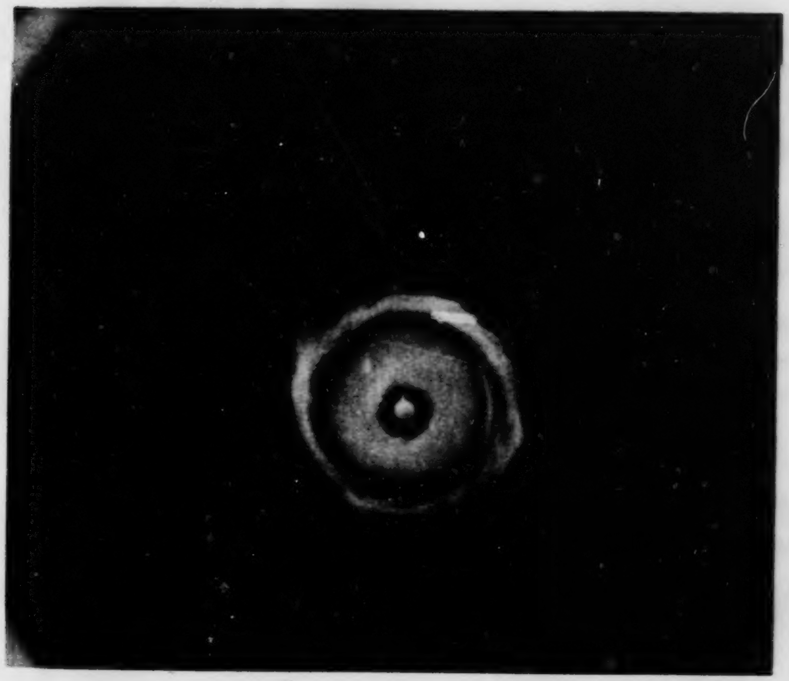


FIGURE 14. FLAME AT 0.25 POUND/HOUR FUEL FLOW

mixing that inevitably occur. At low fuel flows, the most efficient flame resulted when the primary zone was leaned out with excess air. This shortened the flame, reduced heat losses, and hence provided more efficiency.

4) The increase in CO as a function of fuel flow. A not unexpected result in view of the time dependence of the reaction. By extension of the secondary combustor zone, the CO emissions can be reduced and, provided good fuel-air mixing has been obtained, NO emissions theoretically would not increase significantly.

Tests were done with a water cooled heat exchanger mounted on the rear of the combustor and designed to simulate the vaporizer that, in the final design, must be used. NO emissions were significantly reduced, CO increased slightly, but hydrocarbon emissions increased an order of magnitude. It is concluded that heat losses to the vaporizer must therefore be minimized by shielding it from the flame.

Tests were then done using the fan and control system previously shown in Figure 4. The initial combustion characteristics were totally different and emissions unacceptably high. Two main features were noted. At high air flows, circumferential air maldistributions were so high that raw fuel escaped from the exhaust. At low air flows a pronounced swirl occurred in the flame. Air distribution circumferentially was good but axially considerably different from the rig tests. Aerodynamic analysis

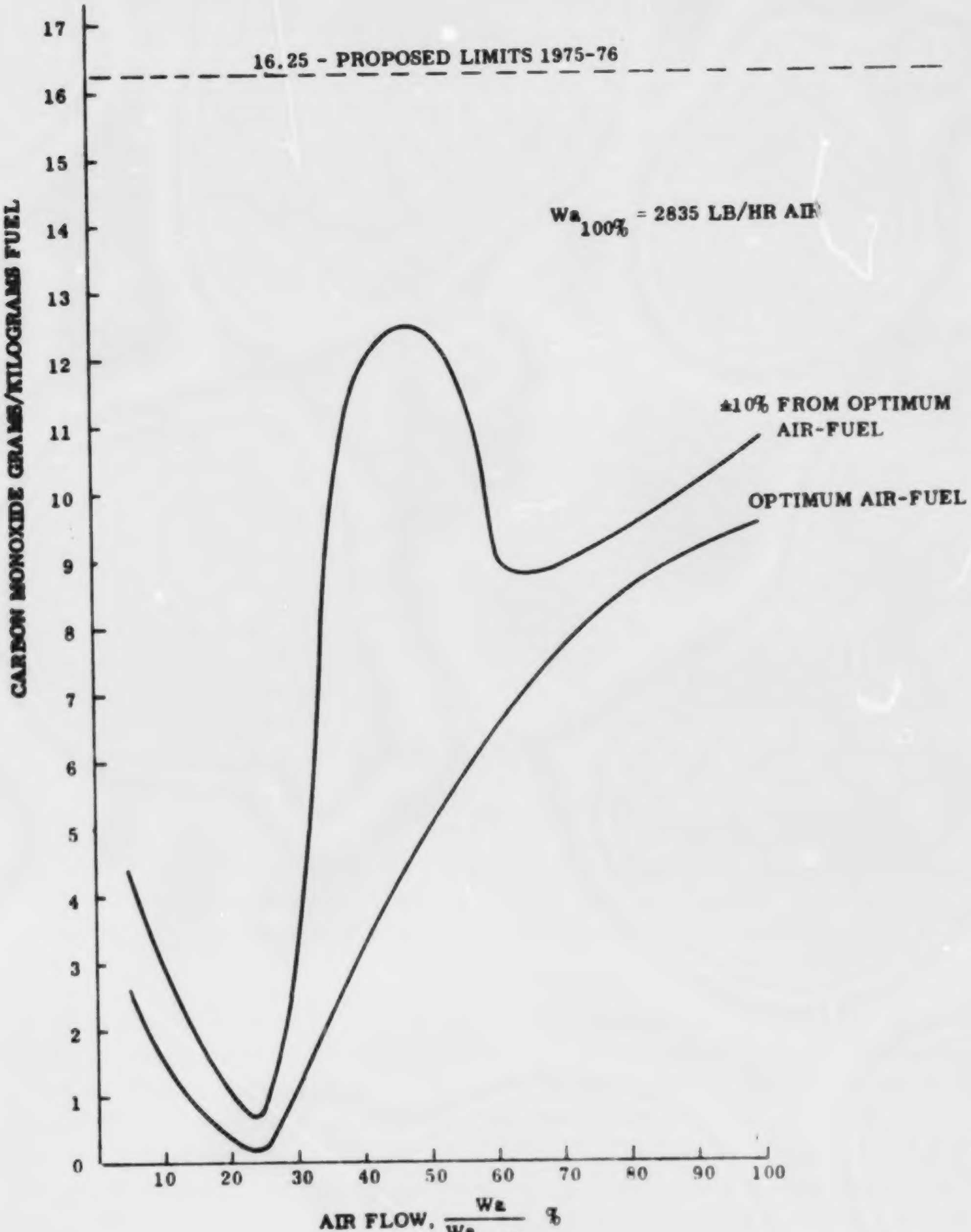


FIGURE 15. EMISSIONS OF CARBON MONOXIDE AS A FUNCTION OF COMBUSTOR AIR FLOW, AT OPTIMUM AIR FUEL FOR MINIMUM EMISSIONS AND ALSO WITH A ±10% DEVIATION OF AIR FUEL FROM OPTIMUM

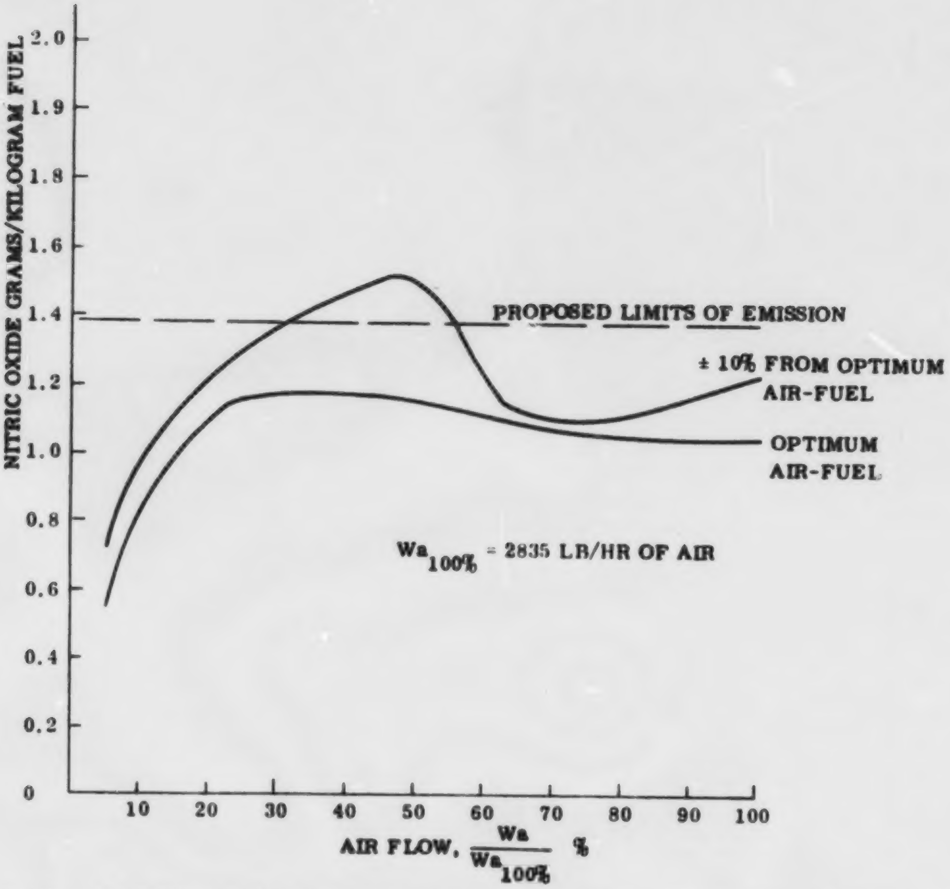


FIGURE 16. EMISSIONS OF NITRIC OXIDE AS A FUNCTION OF COMBUSTOR AIR FLOW, AT OPTIMUM AIR FUEL FOR MINIMUM EMISSIONS AND ALSO WITH A ±10% DEVIATION OF AIR FUEL FROM OPTIMUM

indicated these two problems were caused by two design features in the fan design. The fan used was a highly loaded solid vortex design typical of those used in aircraft where minimum size and weight is needed (Ref. 8). The velocities in it were high and to supply air to the combustor, substantial diffusion was required. Diffusion is unstable and results in circumferential maldistributions. At low air flows, the presence of slight swirl from the fan results in the establishment of a free vortex in the combustor of relatively high velocity (the fan runs continuously at full speed). Because the combustor pressure drop is low (0.0008-inch water at 1.0 pounds per hour of fuel) this has a significant effect on the discharge coefficients of the air metering holes with consequent axial maldistributions. Aerodynamic problems of this sort are the bane of combustion design. In gas turbines, not only is efficiency reduced but serious hot spots are caused. The much wider range of air flows of the Rankine combustor makes

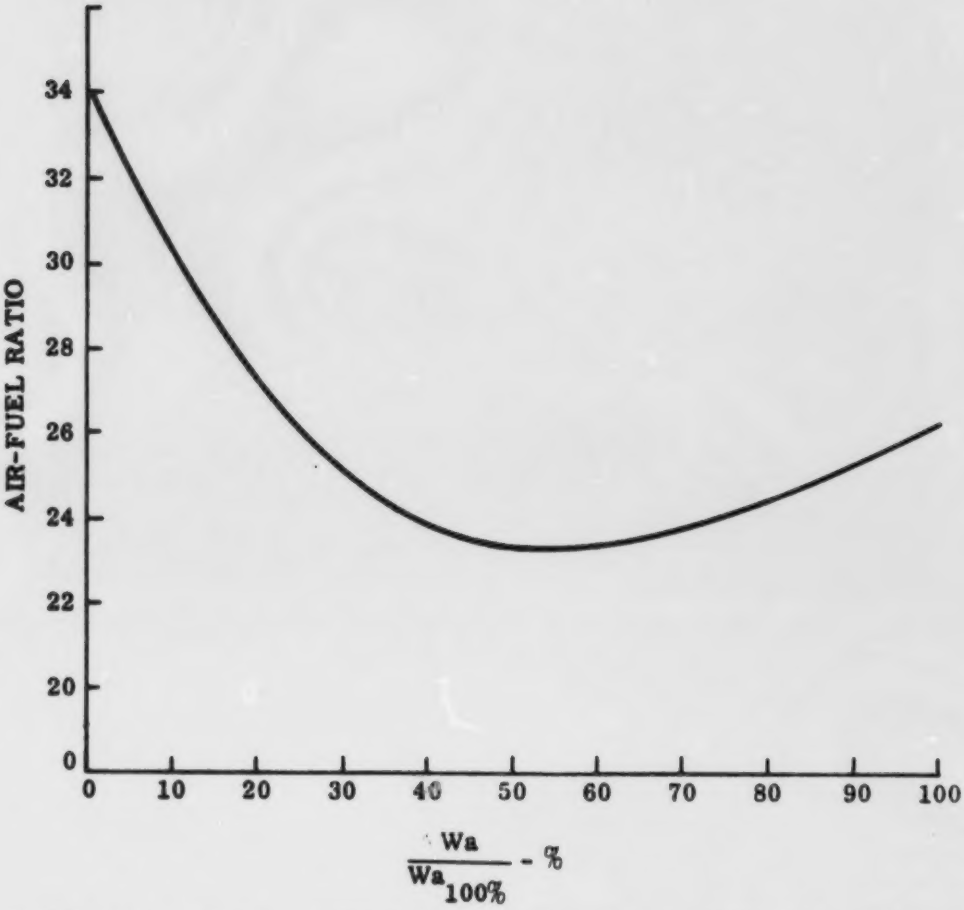


FIGURE 17. AIR-FUEL RATIO FOR MINIMUM EMISSIONS AS A FUNCTION OF COMBUSTOR AIR FLOW

the problem of precise air-fuel control more critical and the tests indicate it to be a leading problem if emissions are to be kept low and vaporizer life to be long.

The expedient of introducing a large plenum between fan and combustor (Fig. 18) provided a temporary solution and rig and fan tests were found to aerodynamically duplicate each other. This was confirmed by exhaust temperature traverses of both rig and fan air supplies and which were in close agreement (Fig. 19). The profile of temperature is indicative of inadequate mixing. Apart from detracting from vaporizer reliability, it is bound to raise emissions because optimum air-fuels are not maintained.

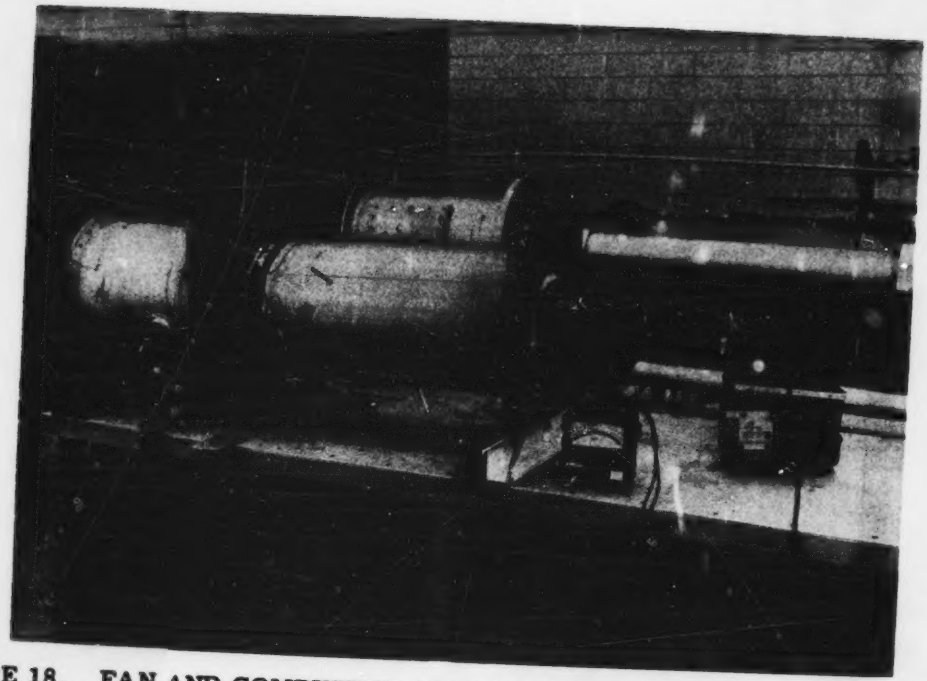


FIGURE 18. FAN AND COMBUSTOR ASSEMBLY UTILIZING LONG MIXING DUCT

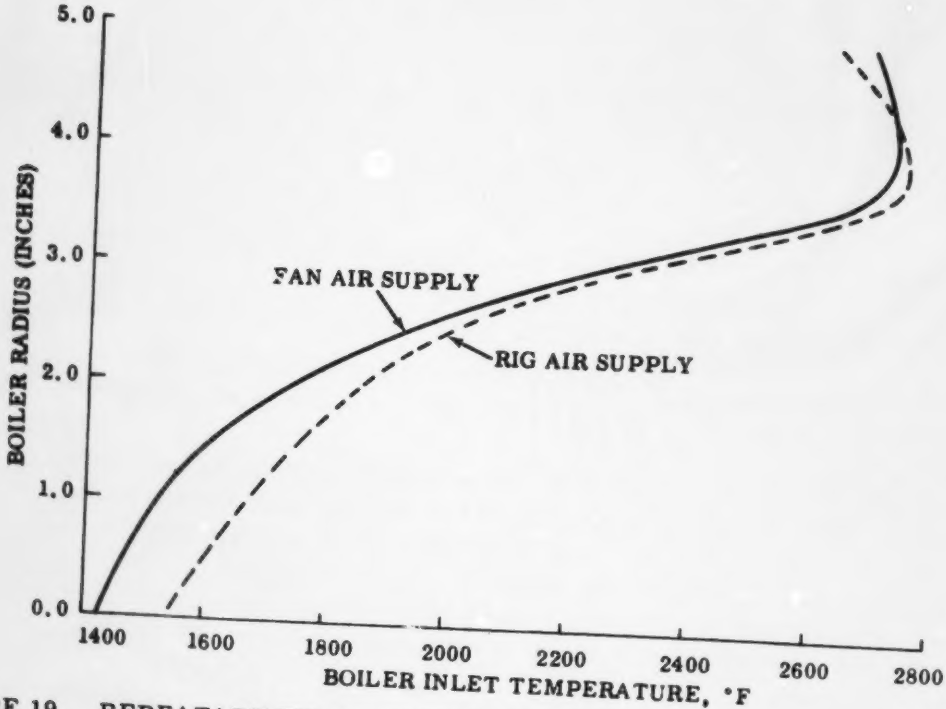


FIGURE 19. REPEATABILITY OF RADIAL PROFILE OF TEMPERATURE INTO BOILER USING BOTH FAN AND RIG AIR SUPPLIES

As the plenum involved a duct of 36 inches in length, it is not acceptable for use in an auto. The solution lies in a fan designed specifically for the combustor, of larger diameter and lower velocity. The air control system would retain the present features and it is doubtful whether any other air control system, such as a butterfly valve, would be satisfactory because of the asymmetry of air that would result.

After further development, endurance tests indicated the most significant problem to be carbon build up in the combustor walls. Temperature and emission measurements, as well as visual flame observation indicate that improvements in air-fuel mixing are needed; and it is expected that further development can provide substantial performance improvements.

5.2 CONTROL SYSTEM DEVELOPMENT TESTS

Control system development tests have been completed on all major components. Final fuel metering valve performance tests have demonstrated the gain to be 61 pounds per hour per inch stroke with a linearity of ±2.5 percent across a range of 3 to 115 pounds per hour (Fig. 20). Good repeatability and flow control to 0.5 pounds per hour indicates the valve has a dynamic range greater than 200 to 1 (Fig. 21). Fuel pressure regulator valve (Fig. 22) development tests have extended its range and improved accuracy. An important goal of extending the control range to 1.0 inch of H_2O input actuator pressure differential has been achieved. Blower motor power reductions of as much as 65 percent can now be compensated with the fuel regulator. Correct fuel flow can be maintained within ±4 percent with actuator input differential pressures ranging from 1.0 to 2.5 inches of H_2O (Fig. 23).

Air metering valve problems caused by high dynamic head, swirl, and turning losses by use of an off-the-shelf blower, required high pressure loss baffles and flow straighteners to be installed. As a consequence, the blower motor was required to be operated approximately 6 volts above its normal input. Additionally, combustor tests with the air valve showed that potential aerodynamic problems existed that would require considerable development effort to minimize volume. Analysis of these problems indicated that the best solution would be to specially design a matched fan-motor combination to the functional and geometric requirements of the combustor. It was established that the demonstration system should utilize a simple dump plenum to make the fan more aerodynamically compatible with the air valve and combustor. Thus, a 36-inch extension between the fan and air valve was incorporated into the demonstration valve. Figure 24 shows typical performance on bench tests of the air metering valve.

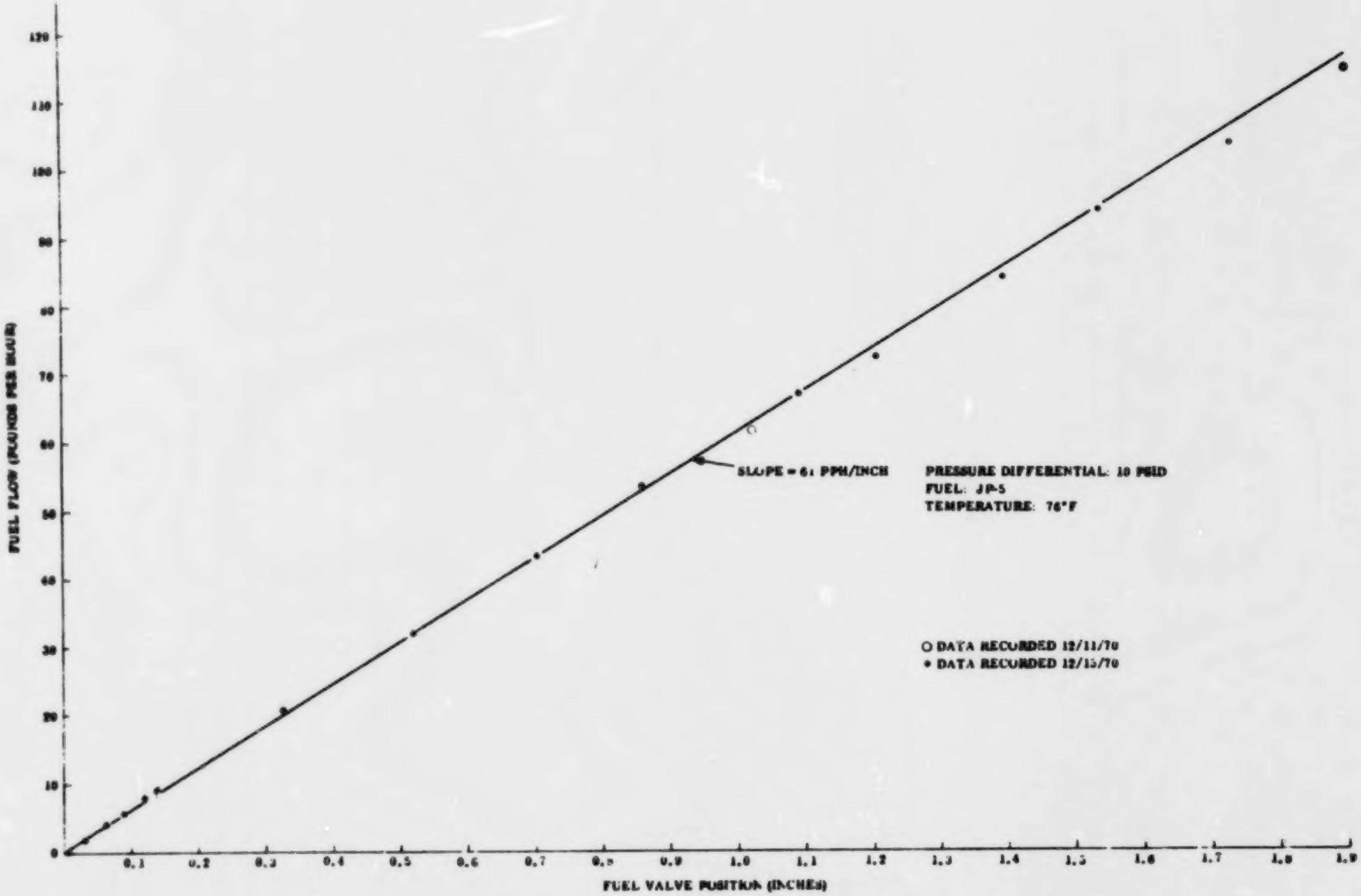


FIGURE 20. FUEL METERING VALVE FINAL WEIGHT FLOW CALIBRATION

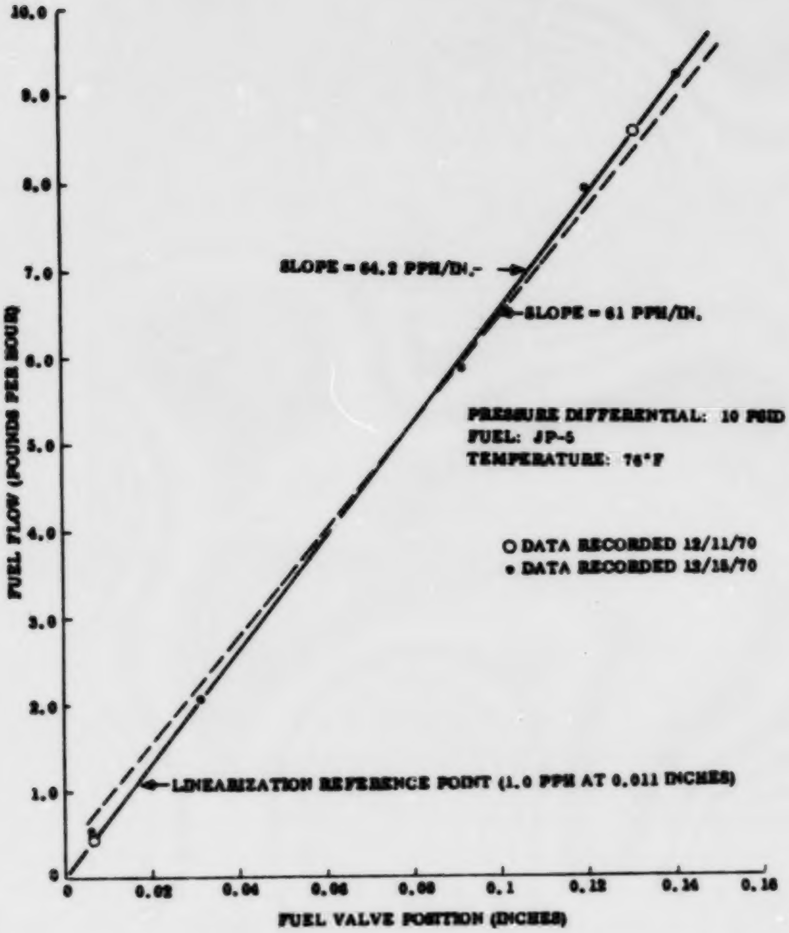


FIGURE 21. FUEL METERING VALVE FROM 0.5 TO 10 POUNDS PER HOUR (FINAL WEIGHT FLOW CALIBRATION)

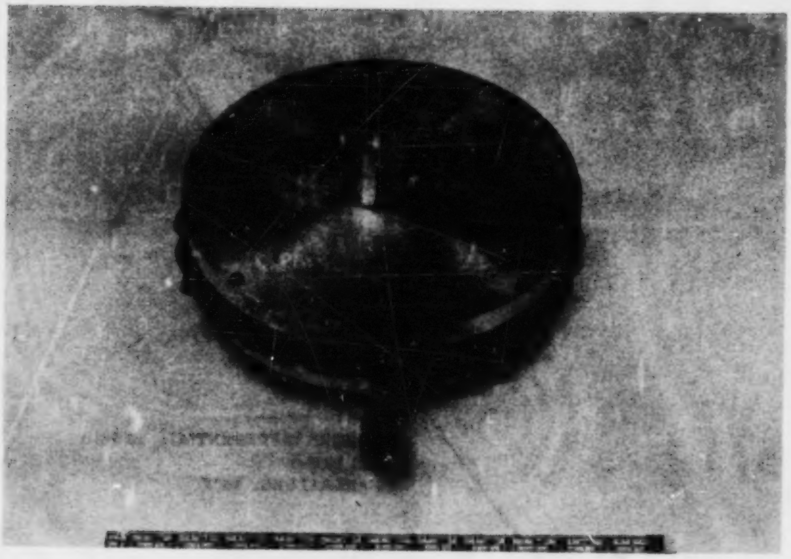


FIGURE 22. FUEL REGULATOR ASSEMBLED WITH $P_1 - P_2$ ACTUATOR

5.3 EMISSION MONITORING

All emission data presented was taken with Beckman Model 315A, CO, CO_2, and NO analyzers along with Beckman Model 402 Hydrocarbon Analyzer. The emission monitoring system is designed in accordance with existing vehicular specifications as defined by Federal Standards, the Automobile Manufacturing Association and the State of California. It was selected for use in this program because its wide spread acceptance permits direct comparison of results with other investigators' work. The CO, CO_2, and NO analyzer uses NDIR cells selected for the ranges required for the program. Table I lists the ranges, repeatability and interference factors that are characteristic of this measurement system. In order to achieve a 0-150 ppm range for NO, an exceptionally long (41 inches) NDIR cell is incorporated for this important parameter. Figure 25 shows the control panel and strip chart data readout employed with these instruments. Sampling was accomplished with a 1/8-inch diameter cooled probe for CO, CO_2, and NO to prevent reaction in the sampling line. Hydrocarbons were sampled with a line maintained at 350° F by means of electric heaters.

VI. CONCLUSIONS

The 1975-76 emission levels for automobiles are feasible when using a Rankine engine combustor. Package size and power requirements of the combustor need not be excessive, the components used are not complex, are capable of low cost mass production and, as involve proven concepts, high reliability is attainable. The fuels used need no critical refining capabilities, and are much safer and less expensive than gasoline. They can be instantly burned in the coldest weather and do not have any

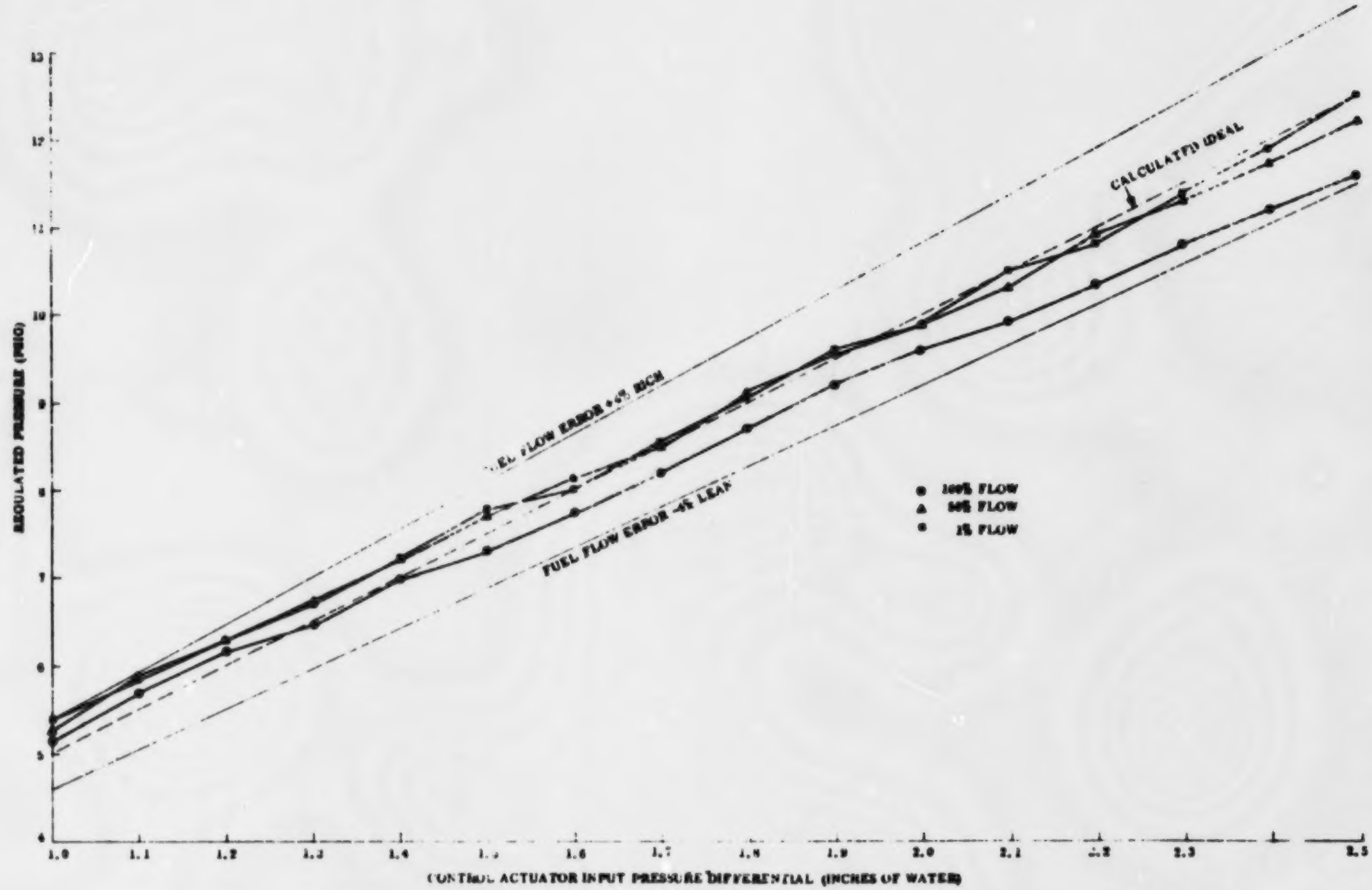


FIGURE 23. REGULATOR CALIBRATION WITH 0.008 INCH FLAT RUBBER DIAPHRAGM

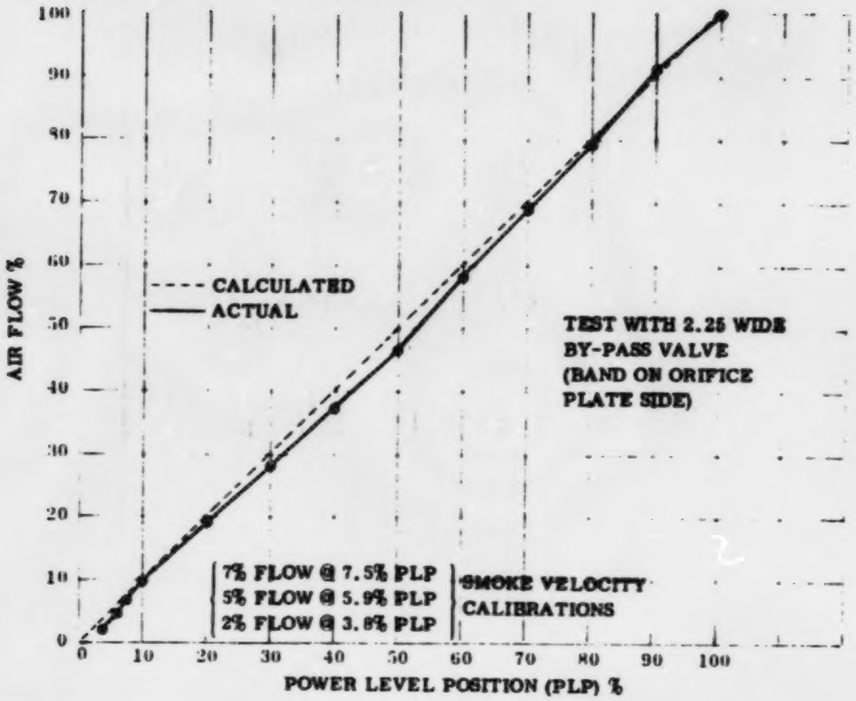


FIGURE 24. FLOW CONTROL PERFORMANCE OF AIR METERING VALVE

TABLE I

INSTRUMENTATION ACCURACY USED FOR DATA ANALYSIS

Constituent Gas	Ranges	Repeatability % of full Scale	Interference			Calibration Gas Accuracy
			Interfering Gas	Mole Percent	Extraneous Response	
CO	0-1000 ppm by Vol. 0-2.5% by Vol. 0-250 ppm by Vol.	1.0 1.0 1.0	CO_2 H_2O	5.0 3.3	3.4 ppm 1.1 ppm	±5%
CO_2	0-16% by Vol. 0-5% by Vol.	1.0 1.0	---	---	---	±5%
NO	0-1000 ppm by Vol. 0-150 ppm by Vol.	±1.0 ±2.0	CO CO_2	5.0 5.0	1 ppm 3 ppm	±5%
H_xC_y	0-5 ppm 16 ranges through 0-250,000 ppm	<±5	---	---	---	±5%

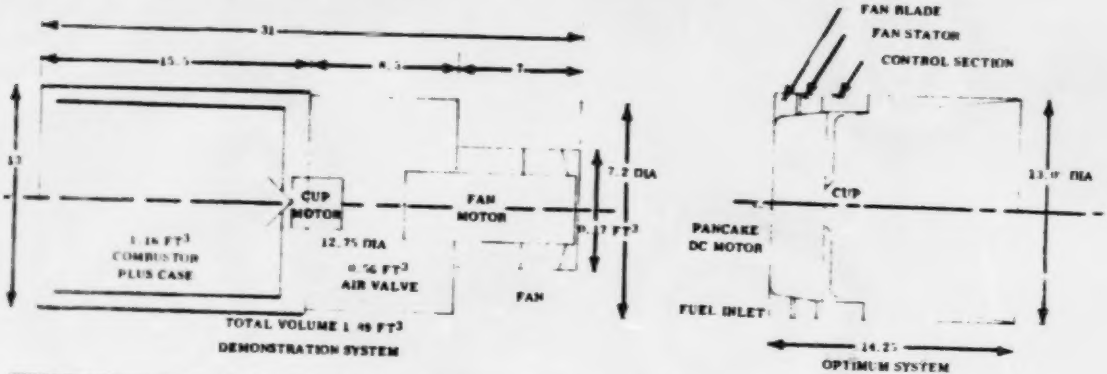
FIGURE 25. TOTAL VIEW OF BECKMAN GAS ANALYZER INCLUDING THE FLAME IONIZATION DETECTOR

warm-up emissions as occurs on current engines. They need no special additives for combustion control, nor will the current automotive evaporative control systems be necessary.

Development is necessary, especially in regard to integration with other engine components because they can affect both the shape of final performance of the combustor and particularly, the critical aerodynamic interface necessary for low emissions. The most critical area is in air-fuel control. To achieve the best possible performance (and the results, while below current goals, could be substantially improved) requires aerothermodynamic sophistications more typical of advanced gas turbine combustors than conventional heating systems.

Further development of the combustor components would yield substantial benefits in both size and power requirements (Fig. 26). However, if an economic vaporizer of reasonable size was used (Ref. 11), the horsepower requirements of the combustor would increase from 1.25 to 3.65. By developing the inherent variable fan speed capability of the fuel air control system, the parasitic losses at part loads could be maintained at approximately 1 horsepower. Typically, maximum power is required only transiently and normal power output is 35 percent or less than maximum. This suggests that the design should be optimized for the normal engine load demands of 35 percent power or less.

If the motor speed was reduced to supply the air required for 35 percent power, the total power required of combustor and vaporizer would be reduced from 3.65 to 1.25 horsepower, and full fan power would only be required in occasional engine power demands above 35 percent where fan speed would be increased. Such a design concept provides considerable potential for combustor improvements as it permits the use of higher combustor pressure losses and a smaller combustor.



	Demonstration System Design	Optimum System Design	Spec
Total Volume (ft^3)	1.89	1.09	1.33
Length (Inch)	31.0	14.25	
Diameter (Inch)	13.00	13.00	
Combustor Volume (ft^3)	0.687	0.687	
Combustor Diameter (Inch)	11.00	12.00	
Horsepower	2.30	1.25	2.00
Combustor Loss (Inch Water)	8.00	6.0	
Diffuser Loss (Inch Water)	3.00	0.5	
Metering Loss (Inch Water)	2.00	1.5	
Overall Pressure Loss (Inch Water)	13.00	8.0	
Fan Efficiency (%)	75.0/	85.0	
Motor Efficiency (%)	75.0	75.0	
Overall Efficiency (%)	56.2	68.0	
Motor Speed (rpm)	14000	7000	

Note: Ignition, Fuel and Air Regulators not included.

FIGURE 26. TWO FAN SKETCHES - PRESENT AND OPTIMUM

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Chapter XI

HYBRID HEAT ENGINE/ELECTRIC SYSTEMS STUDY

by

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This briefing summarizes the results of a comprehensive broadbased study aimed at determining the feasibility of using a hybrid heat engine/ electric propulsion system as a means of reducing exhaust emissions from street-operated vehicles. In this hybrid concept, the source of power is a combination of heat engine and batteries (in essence, the heat engine supplies steady state power and the batteries supply transient power demands). The study examined--for several classes of vehicles--many types of heat engines, batteries, and other major components, as well as several design configurations. Following a review of the associated technologies, hybrid performance, exhaust emissions, and major component requirements were determined. Based on these results, recommendations can be formulated to ensure the development of critical powertrain components for an early demonstration of prototype vehicles.

In the propulsion of the hybrid heat engine/electric vehicle, the ultimate source of all energy to be expended is the heat engine. The key to success in reducing exhaust emissions is good part-load and full load efficiency of powertrain components, and the ability to restrict operational requirements for the heat engine to those of supplying road load power and (in conjunction with a generator) recharging advanced high power/high energy batteries that supply acceleration power. With this idea in mind, the study was tailored to examine six classes of vehicles: the 4000-lb family car, 1700-lb commuter car, low- and high-speed postal/delivery van, and low- and high-speed intracity bus. For each class of vehicle, five engines were included in the powertrain: spark ignition, compression ignition, gas turbine, Rankine cycle, and Stirling cycle. Lead-acid, nickel-cadmium, and nickel-zinc batteries were studied for adequacy in supplying acceleration power to each vehicle. Also, a wide range of ac and dc motors, generators, and power conditioning and control systems were evaluated for performance efficiency, weight, simplicity, and cost.

Throughout the study, the following ground rules prevailed:

- The hybrid vehicle should match the conventional automotive vehicle in acceleration, speed, gradeability, curb weight, and powertrain weight.
- External recharge of the battery should not be required. This requirement was simulated in computations by requiring that the heat engine-driven generator recharge the battery to the original state-of-charge prior to the end of a selected emission driving cycle.
- The battery is to discharge only when the vehicle is undergoing acceleration, not on a smooth grade or at cruise conditions.
- The heat engine is to supply steady road load power and is not required to undergo rapid acceleration.
- Only design concepts compatible with near term (1972-1975) prototype vehicle development are to be considered.

The following set of charts summarizes the study content, parameters examined, and the results. Some of the results are highlighted in the text below for the family and commuter car.

- Only the spark ignition internal combustion engine and the gas turbine engines can be practically packaged into the hybrid heat engine/electric vehicle with the performance specified in the study.
- All hybrids examined showed marked emission reductions over current conventional vehicles.
- If currently available technology--not including catalytic converters--is used, no version of the family car could meet 1975 emission standards (HC = 0.46 grams/mile, CO = 4.79 grams/mile and NO_2 = 0.4 grams/mile).
- If advanced technology--including the catalytic converter for the internal combustion engine--is used, all versions but the diesel could meet 1975 standards (except for the minor NO_2 excess for the spark ignition family car version).
- Commuter car emissions are less than one-half of those for the family car and with advanced technology easily meet the 1975 standards. (The commuter car weighs only 1700 lb and has reduced acceleration and maximum cruise speed capabilities.)

- Emissions are approximately 10 and 15 percent lower for the parallel powertrain configuration as compared to the series configuration in the family and commuter cars, respectively.
- Study results are based on hot start data. Incorporation of cold start effects would still allow the advanced technology versions of some hybrid vehicles to meet 1975 standards.
- Regenerative braking has essentially no effect on emissions.
- Motor and engine part-load characteristics and motor efficiency are extremely important in determining emissions.
- Improved lead-acid batteries could be used in near term hybrids. Increased energy density and power density capabilities are needed. Battery lifetime and improved charge acceptance are the most critical areas requiring improvement. Nickel-zinc looks promising for the post-1975 period.
- Battery charge acceptance characteristics play an extremely important role in determining resultant vehicle exhaust emissions.
- Hybrid fuel consumption is about the same as for the current conventional car.
- For the hybrid using a spark ignition engine, high-production costs would range from 1.4-1.6 times todays car.

PURPOSE OF STUDY

- TO DETERMINE :
  - FEASIBILITY OF HEAT ENGINE/ELECTRIC HYBRID AUTOMOTIVE VEHICLES
  - POTENTIAL REDUCTION IN AUTOMOTIVE EXHAUST EMISSIONS
  - MOST PROMISING DESIGN CONCEPT IN EACH VEHICLE CLASS :
    - FULL-SIZE FAMILY CAR
    - SMALL COMMUTER CAR
    - DELIVERY AND POSTAL VAN
    - CITY BUS
- TO RECOMMEND :
  - TECHNICAL DEVELOPMENT PLAN FOR CRITICAL COMPONENTS
  - TECHNICAL DEVELOPMENT PLAN TO ASSURE PRODUCTION VEHICLE IN 1975 - 1980 PERIOD
    - SCHEDULE
    - RESOURCES ALLOCATION
    - MILESTONES (e.g. TEST BEDS, PROTOTYPES ···)

THE HYBRID CONCEPT

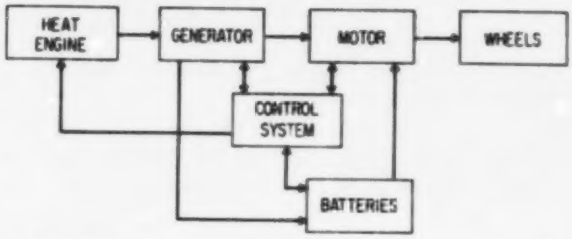
- POWER FOR PROPULSION IS SUPPLIED BY TWO SOURCES : HEAT ENGINE AND BATTERIES
- POWER FOR ACCELERATION IS SUPPLIED BY THE BATTERIES - CRUISE POWER IS SUPPLIED BY THE HEAT ENGINE
- HEAT ENGINE SIZE CAN BE REDUCED
- HEAT ENGINE OPERATES OVER RESTRICTED RPM RANGE
- ENGINE DESIGN AND OPERATION CAN BE OPTIMIZED TO REDUCE EXHAUST EMISSIONS
- ALLOWS INTERMEDIATE STEP BETWEEN CURRENT INTERNAL COMBUSTION AND PRACTICAL ALL-ELECTRIC VEHICLE OF 1985-1990

ADVANTAGES OF OPERATING HEAT ENGINE IN HYBRID VEHICLE

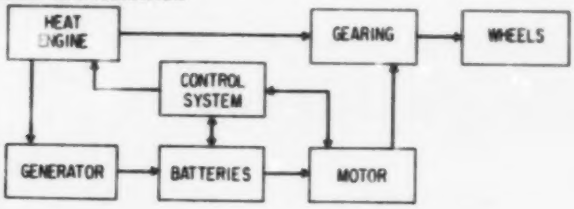
- RAPID ENGINE ACCELERATION REQUIREMENT IS REMOVED
  - EXPECT NO STUMBLE FROM LEAN ENGINE OPERATION
  - EXPECT IMPROVED EXHAUST EMISSIONS
- RPM AND LOAD RANGE ARE RESTRICTED
  - EXPECT IMPROVED COMBUSTION AND FUEL CONSUMPTION
  - EXPECT IMPROVED EXHAUST EMISSIONS
  - REDUCE DESIGN REQUIREMENTS FOR CATALYTIC CONVERTER

SCHEMATIC OF HYBRID CARS

SERIES CONFIGURATION



PARALLEL CONFIGURATION



VEHICLE COMPONENT ARRAY

ENGINES
- I.C. SPARK
- DIESEL
- GAS TURBINE
- RANKINE CYCLE
- STIRLING CYCLE

BATTERIES
- LEAD-ACID
- NICKEL-CADMIUM
- NICKEL-ZINC

MOTORS
- A.C. INDUCTION
- D.C. SHUNT WOUND/ EXTERNALLY EXCITED
- D.C. COMPOUND
- D.C. SERIES
- D.C. BRUSHLESS

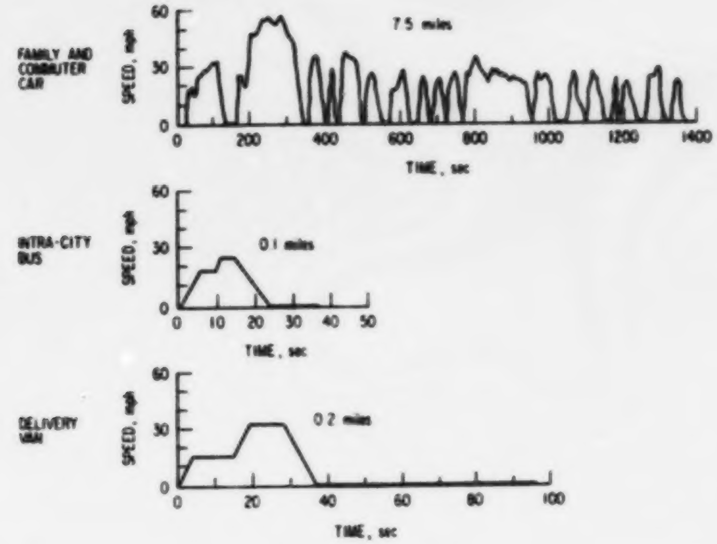
GENERATORS
- D.C.
- A.C. (ALTERNATOR)

POWER CONDITIONING AND CONTROL
- SILICON CONTROLLED RECTIFIERS
- INVERTERS
- SOLID STATE INTEGRATED CIRCUITS
- CYCLOCONVERTER
- RELAYS/SWITCHES
- RESISTORS/INDUCTORS

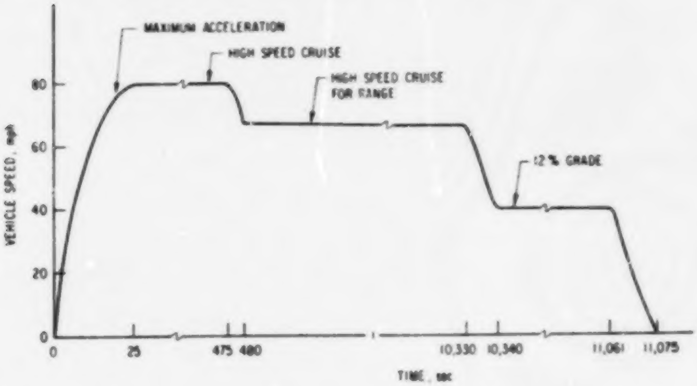
HYBRID VEHICLE SPECIFICATIONS

	FAMILY CAR	COMMUTER CAR	INTRA - CITY BUS	DELIVERY /POSTAL VAN
V MAX (miles/hr)	80	70	40	40
V GRADE AT GRADE (miles/hr, AT %)	40 AT 12	33 AT 12	6 AT 20	8 AT 20
RANGE (miles)	200	50	200	60
LOADED WEIGHT (lb)	4,000	1,700	30,000	7,000
ASSIGNED POWER TRAIN WEIGHT (lb)	1,500	600	6,000	1,700
ASSIGNED POWER TRAIN VOLUME (ft)	28	16	175	42
ACCELERATION	EQUAL TO CONTEMPORARY AUTOMOTIVE VEHICLE			

DRIVING CYCLES FOR EMISSION COMPARISONS

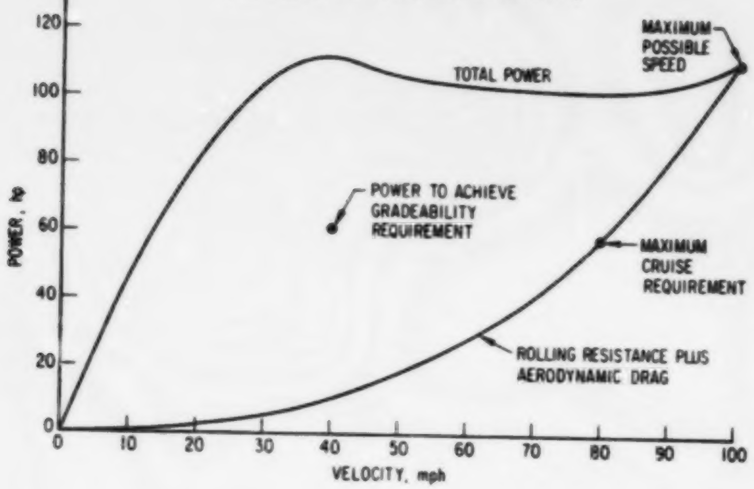


DESIGN DRIVING CYCLE, 4000-lb FAMILY CAR

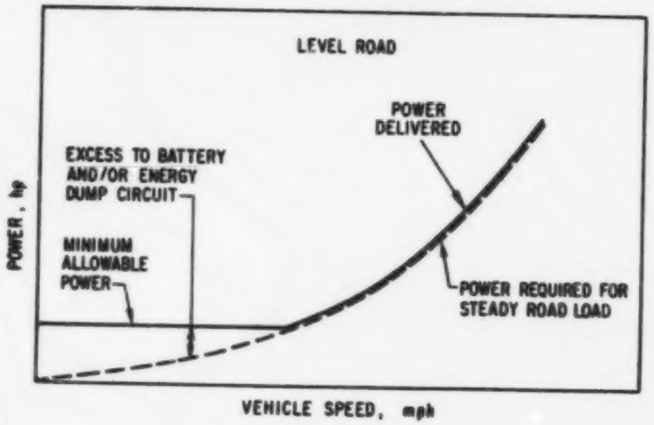


POWER REQUIREMENTS FOR 4000-lb FAMILY CAR AT MAXIMUM ACCELERATION

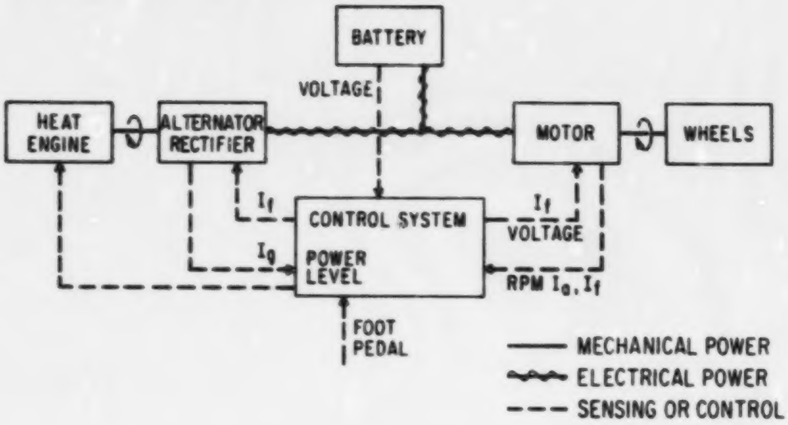
(AT THE POWER CONVERTER OUTPUT)



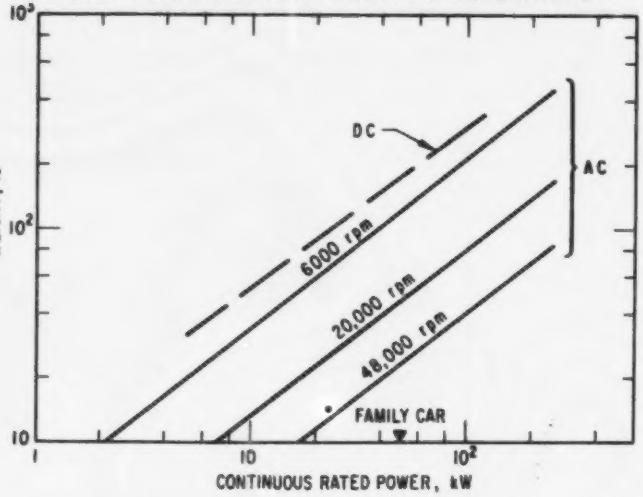
SERIES CONFIGURATION - VARIATION OF HEAT ENGINE POWER WITH VEHICLE SPEED



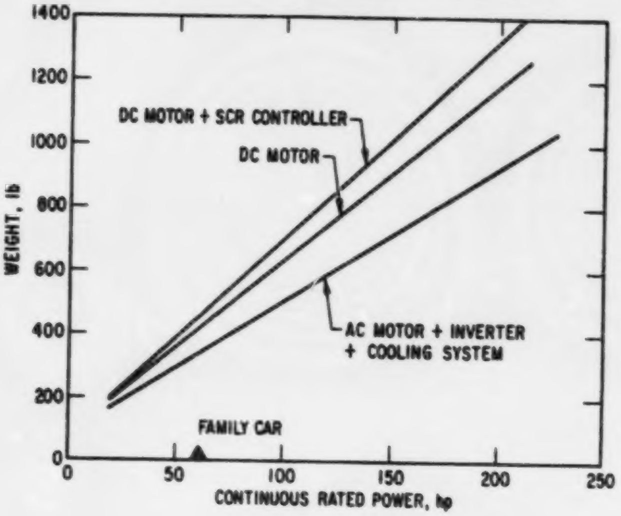
ELECTRICAL CONTROL SCHEMATIC, SERIES CONFIGURATION



WEIGHT COMPARISON FOR ELECTRIC GENERATORS



WEIGHT COMPARISON FOR ELECTRIC MOTORS



HYBRID VEHICLE PERFORMANCE EVALUATION
DIGITAL COMPUTER PROGRAM

CASE INPUT

- HEAT ENGINES EMISSIONS
- VEHICLE WEIGHT
- VEHICLE VELOCITY HISTORY
- BATTERY CAPACITY
- BATTERY RECHARGE EFFICIENCY
- REGENERATIVE BRAKING EFFICIENCY
- TRANSMISSION EFFICIENCY
- [illegible] TRAIN EFFICIENCY
- [illegible] RATIO
- [illegible] GENERATOR CURRENT

ACCOUNTABILITY

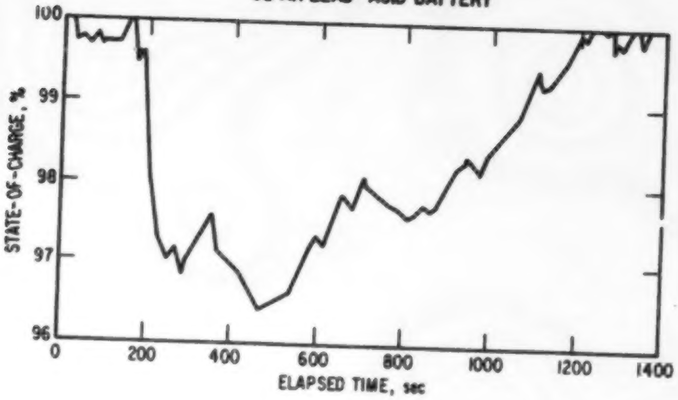
- HEAT ENGINE POWER
- AUXILLARY POWER
- GENERATOR EFFICIENCY
- SYSTEM VOLTAGE
- MOTOR EFFICIENCY
- MOTOR SPEED
- MOTOR TORQUE
- VEHICLE ACCELERATION
- VEHICLE VELOCITY
- TRACTIVE EFFORT
- STEADY ROAD HORSEPOWER
- VEHICLE KINETIC ENERGY
- ROAD GRADE
- AERODYNAMIC DRAG
- TIRE PRESSURE
- TIRE ROLLING RADIUS
- BATTERY CHARACTERISTICS (INCLUDING MAX & MIN ALLOWABLE VOLTAGE)

OUTPUT

- VEHICLE EMISSIONS
- BATTERY STATE-OF-CHARGE
- BATTERY CURRENT
- BATTERY VOLTAGE
- GENERATOR CURRENT
- MOTOR CURRENT
- ENERGY TO ROAD
- ENERGY TO BATTERY
- ENERGY DISSIPATED
- TIME
- DISTANCE TRAVELLED

BATTERY DISCHARGE CHARACTERISTICS DURING DHEW URBAN DRIVING CYCLE

4000-lb FAMILY CAR
GENERATOR OUTPUT = 38 Amp
38 AH LEAD-ACID BATTERY



LEAD-ACID BATTERY DEVELOPMENT GOALS

4000-lb FAMILY CAR

DHEW EMISSION DRIVING CYCLE

38 AH BATTERY OPERATING FOR 1370 sec

7.5 VEHICLE MILES WITH 73 BATTERY CHARGE / DISCHARGE CYCLES AND 1.34 AH DEPTH OF DISCHARGE

DESIGN DRIVING CYCLE

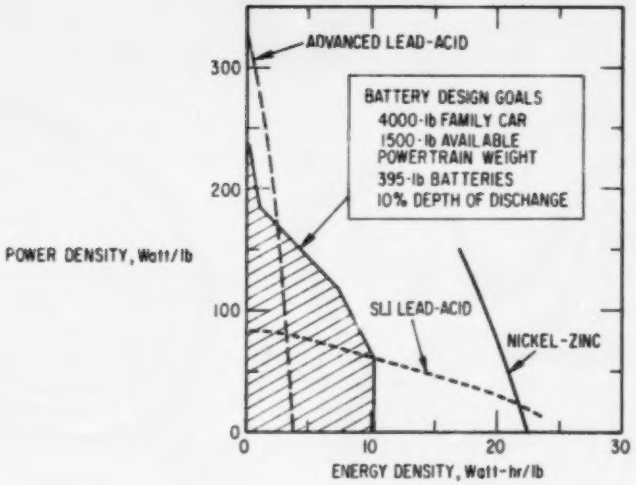
38 AH BATTERY DELIVERED 462 Amp

HYBRID VEHICLE – FAMILY CAR DEVELOPMENT GOALS

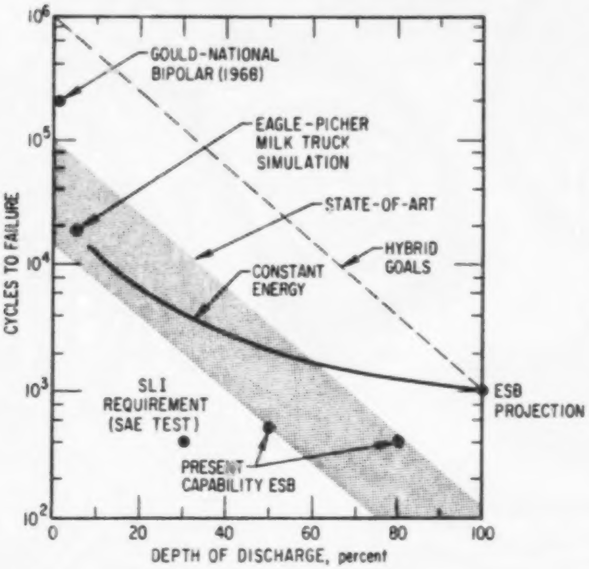
38 AH BATTERY AT LESS THAN 5% DEPTH OF DISCHARGE TO DELIVER UP TO 500 Amp

5000 hr OF OPERATION AND 100,000 VEHICLE MILES WITH 975,000 CHARGE / DISCHARGE CYCLES

COMPARISON OF ENERGY/POWER DENSITY CHARACTERISTICS OF LEAD-ACID AND NICKEL-ZINC BATTERIES WITH DESIGN GOALS



CYCLE LIFE OF LEAD-ACID BATTERIES



CHARACTERISTICS OF SECONDARY BATTERIES

CHARACTERISTIC	BATTERY TYPE		
	LEAD-ACID	NICKEL-CADMIUM	NICKEL-ZINC
RELATIVE COST*	1.0	11.1	2.5
DEMONSTRATED CAPABILITY**			
POWER DENSITY, W/lb	328	450	180
ENERGY DENSITY, W-hr/lb	23.3	18	22
CYCLE LIFE AT DEPTH OF DISCHARGE	137,400 at 6.7%	34,000 at 25%	190 at 100%

*BASED ON ACTIVE MATERIAL COST

**POWER AND ENERGY DENSITIES NOT DEMONSTRATED SIMULTANEOUSLY

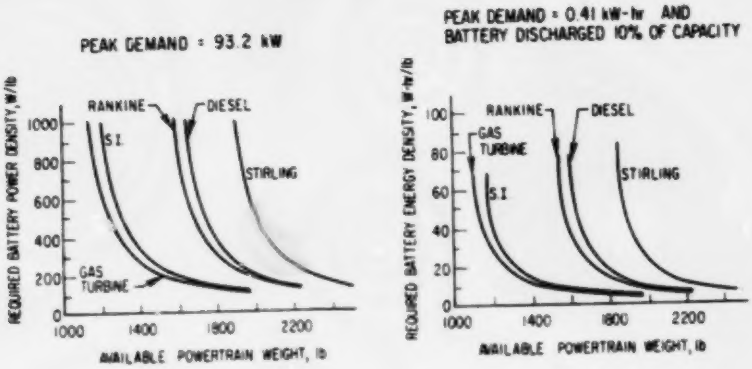
COMPARISON OF HEAT ENGINE CHARACTERISTICS

(89 HP ENGINE)

	WEIGHT NUMBER	VOLUME FT^3	SFC #/BHP-HR
GASOLINE	320	11.4	.50
DIESEL	756	17.3	.40
GAS TURBINE	294	9.1	.57
RANKINE CYCLE	802	12.7	.85
STIRLING CYCLE	1090	21.6	.42

EFFECT OF AVAILABLE POWERTRAIN WEIGHT ON BATTERY DESIGN GOALS

4000-lb FAMILY CAR

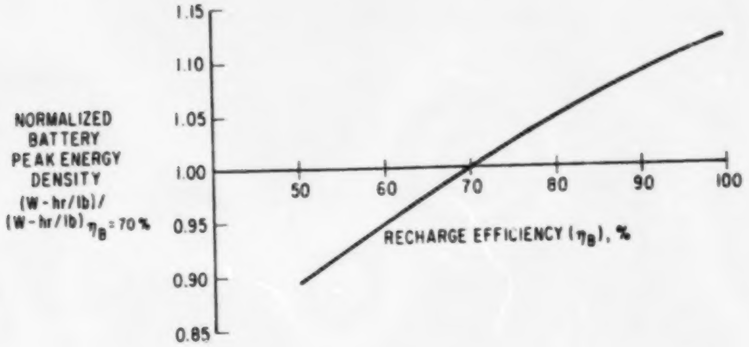


EFFECT OF BATTERY RECHARGE EFFICIENCY ON BATTERY DESIGN GOALS

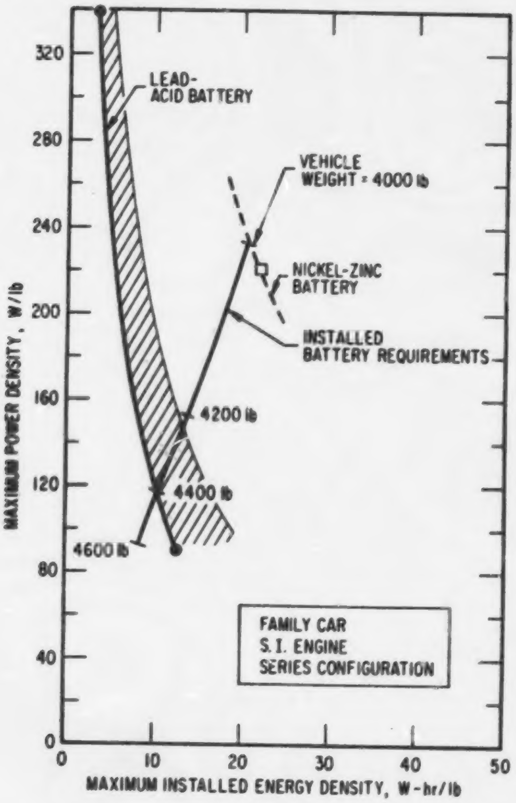
4000-lb FAMILY CAR

FINAL STATE-OF-CHARGE = INITIAL STATE-OF-CHARGE

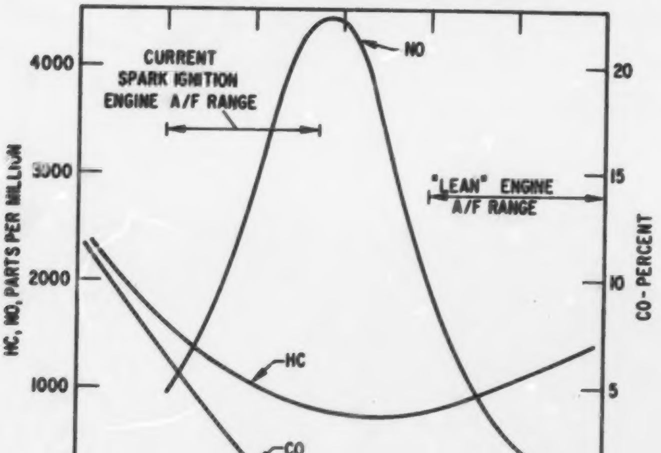
AT $\eta_B = 0.7$, PEAK ENERGY DENSITY = 10.4 W-hr/lb



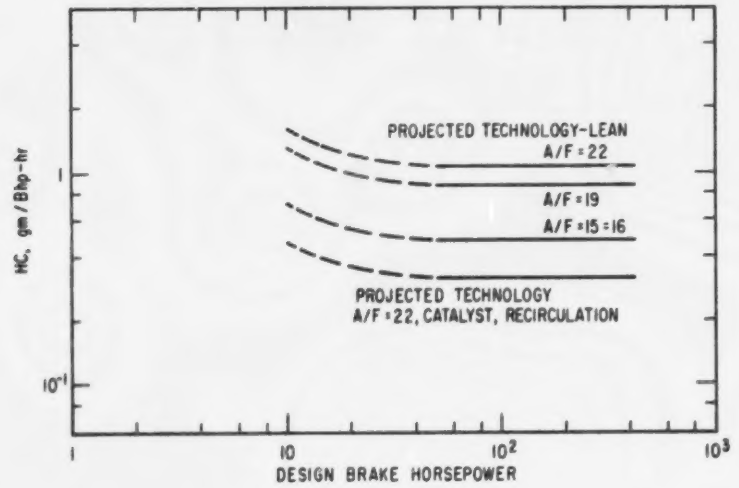
INSTALLED BATTERY REQUIREMENTS AND PROJECTED BATTERY CAPABILITIES



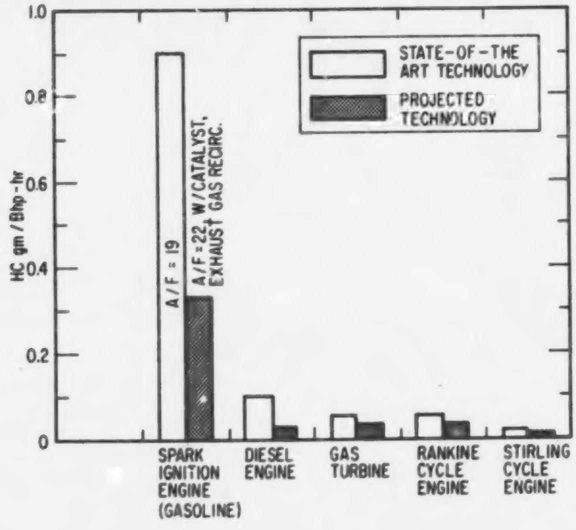
TYPICAL SPARK IGNITION ENGINE EXHAUST EMISSIONS vs AIR/FUEL RATIO (GASOLINE)



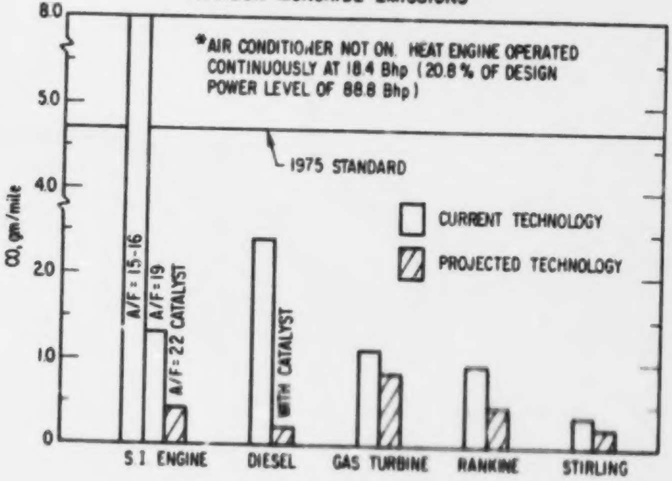
SPARK IGNITION ENGINES (GASOLINE) HYDROCARBON EMISSION, STEADY STATE DESIGN LOAD



HEAT ENGINE EXHAUST EMISSIONS, HYDROCARBON-STEADY STATE DESIGN LOAD, LARGE ENGINES (> 50 hp)

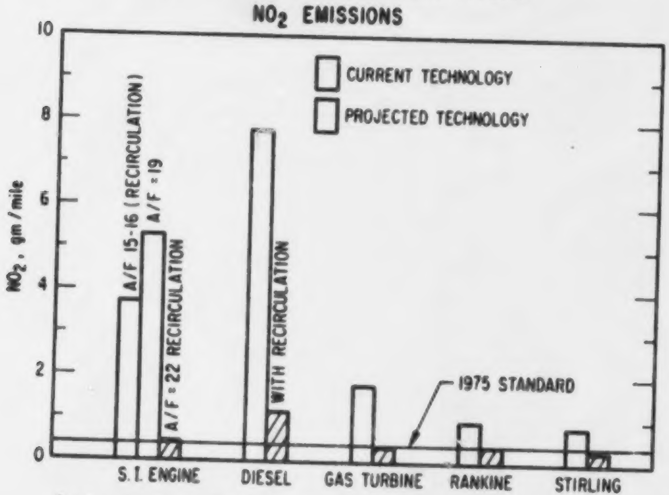


HYBRID (HEAT ENGINE / ELECTRIC) VEHICLE EMISSIONS SUMMARY
4000-lb FAMILY CAR/DHEW CYCLE*
CARBON MONOXIDE EMISSIONS



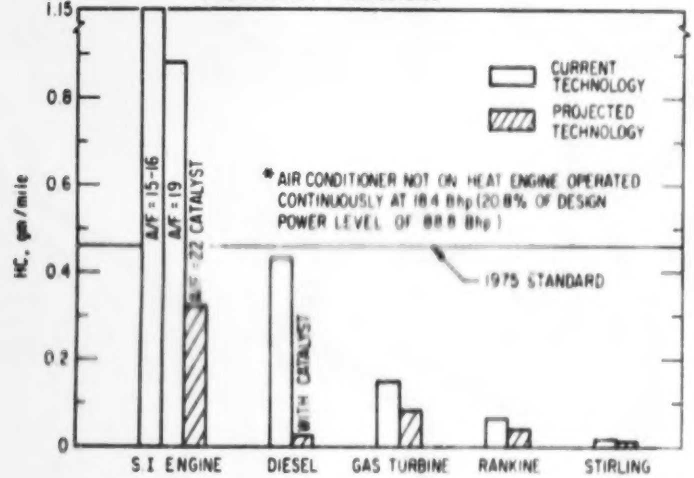
*AIR CONDITIONER NOT ON. HEAT ENGINE OPERATED CONTINUOUSLY AT 18.4 Bhp (20.8 % OF DESIGN POWER LEVEL OF 88.8 Bhp)

HYBRID (HEAT ENGINE / ELECTRIC) VEHICLE EMISSIONS SUMMARY
4000-lb FAMILY CAR/DHEW CYCLE*
NO_2 EMISSIONS

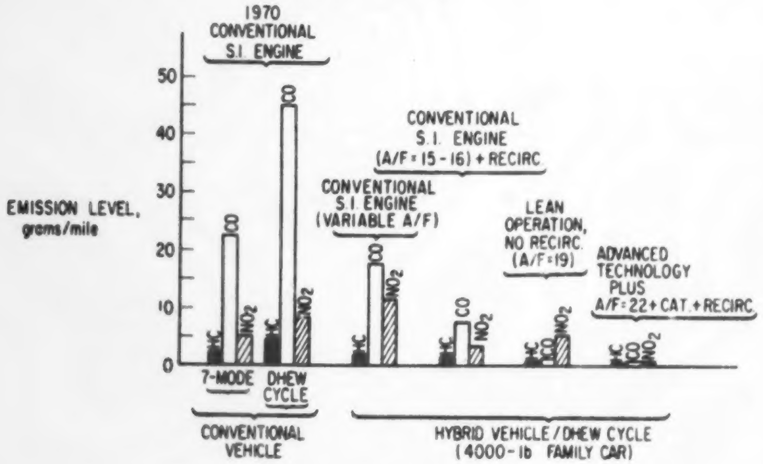


*AIR CONDITIONER NOT ON. HEAT ENGINE OPERATED CONTINUOUSLY AT 18.4 Bhp (20.8 % OF DESIGN POWER LEVEL OF 88.8 Bhp)

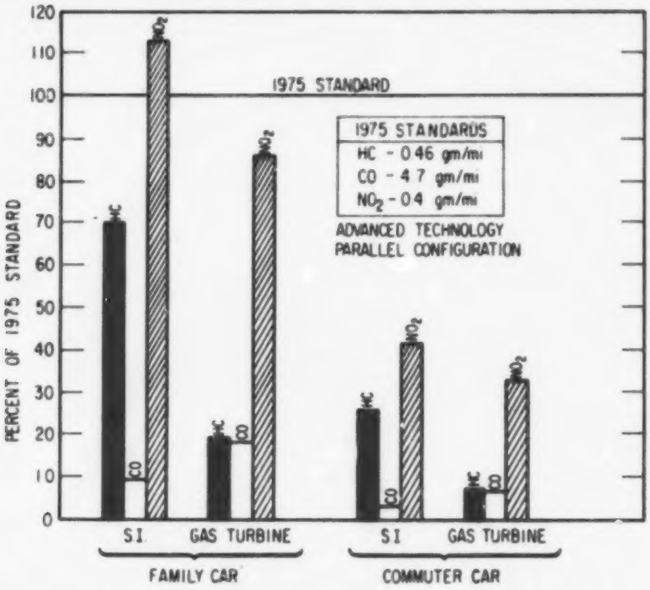
HYBRID (HEAT ENGINE/ELECTRIC) VEHICLE EMISSIONS SUMMARY
4000-lb FAMILY CAR/DHEW CYCLE*
HYDROCARBON EMISSIONS



VEHICLE EMISSION COMPARISON
CONVENTIONAL OPERATION vs HYBRID OPERATION
SPARK-IGNITION ENGINE

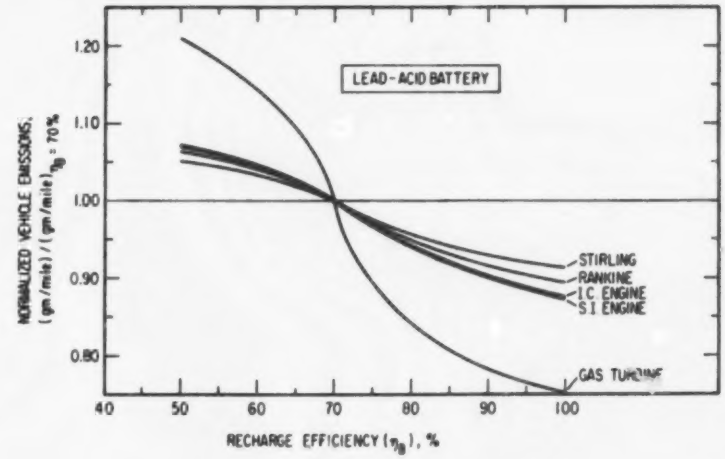


COMPARATIVE EMISSION LEVELS OF THE FAMILY AND COMMUTER CAR

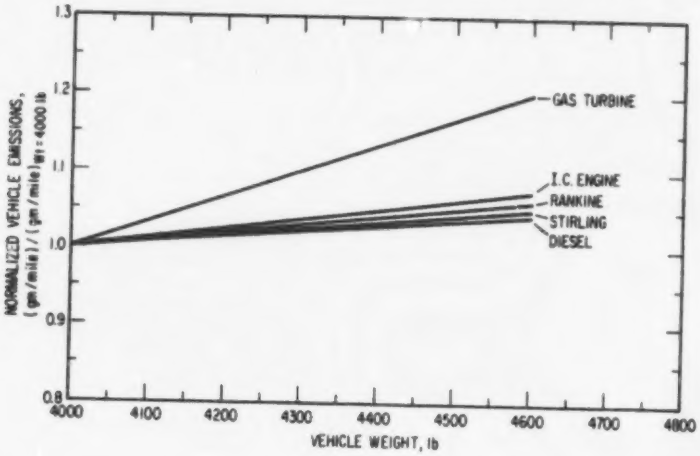


EFFECT OF BATTERY RECHARGE EFFICIENCY ON NO_2 EMISSIONS

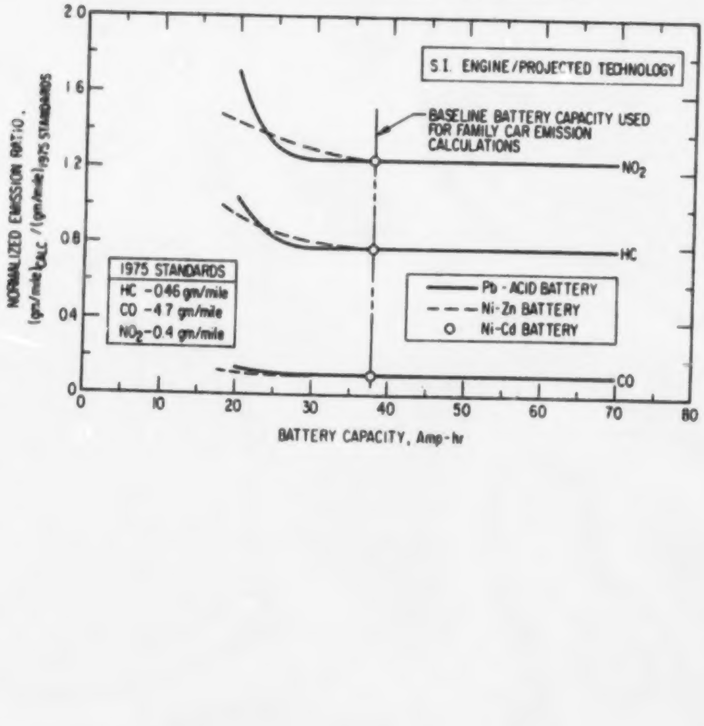
FAMILY CAR - SERIES CONFIGURATION - PROJECTED TECHNOLOGY



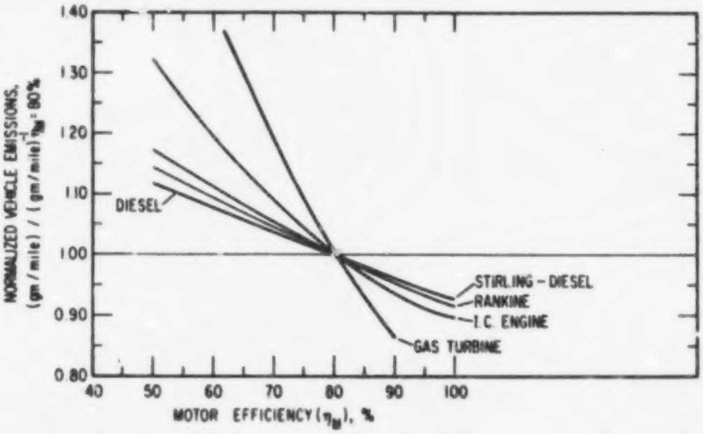
EFFECT OF VEHICLE WEIGHT ON NO_2 EMISSIONS
FAMILY CAR / DHEW CYCLE - PARALLEL CONFIGURATION - CURRENT TECHNOLOGY



EFFECT OF BATTERY CAPACITY AND TYPE ON HC, CO, AND NO_2 EMISSIONS
FAMILY CAR / DHEW CYCLE - SERIES CONFIGURATION



EFFECT OF DRIVE MOTOR EFFICIENCY ON NO_2 EMISSIONS
FAMILY CAR/DHEW CYCLE - PARALLEL CONFIGURATION - CURRENT TECHNOLOGY



HYBRID VEHICLE FUEL CONSUMPTION AND PROJECTED PRODUCTION COSTS

FUEL CONSUMPTION

VEHICLE	SERIES CONFIG (mi/gal)	PARALLEL CONFIG (mi/gal)
COMMUTER CAR	26	30.5
FAMILY CAR	11	12.5
LOW SPEED VAN	3.75	-
HIGH SPEED VAN	4	5
LOW SPEED BUS	1.25	-
HIGH SPEED BUS	1.50	2

HIGH-PRODUCTION COSTS COMPARED TO CONVENTIONAL CAR

VEHICLE	RELATIVE COSTS
CURRENT CONVENTIONAL CAR	1
HYBRID CAR	
SPARK IGNITION	1.4 - 1.6
DIESEL	1.5 - 1.7
GAS TURBINE	~1.6
RANKINE	2+
STIRLING	2.25+

SUMMARY

- HYBRID HEAT ENGINE/ELECTRIC FAMILY CAR
  - SHOWS MARKED REDUCTION IN EMISSIONS OVER TODAY'S CONVENTIONAL HEAT ENGINE-DRIVEN AUTOMOBILE
  - FOR NEAR-TERM DEVELOPMENT, THE I.C. SPARK IGNITION ENGINES OFFER MEANS FOR MEETING 1975 STANDARDS BY PERMITTING LEAN OPERATION
  - FOR THE FUTURE, THE GAS TURBINE CAN EXCEED 1975 STANDARDS FOR ALL EMISSIONS
  - FOR THE FUTURE, RANKINE CYCLE AND STIRLING CYCLE CAN FAR EXCEED 1975 STANDARDS, BUT WEIGHT AND VOLUME CONSIDERATIONS MAY LIMIT THEIR USE. EXTENSIVE DEVELOPMENT ACTIVITY IS REQUIRED
  - SIGNIFICANT IMPROVEMENTS IN ELECTRIC MOTOR PERFORMANCE APPEAR TO BE READILY ACHIEVABLE
  - LEAD-ACID BATTERY TECHNOLOGY IS AVAILABLE (BUT NOT BEING PRODUCED) WHICH CAN SATISFY HYBRID NEEDS FOR POWER AND ENERGY DENSITY REQUIREMENTS BUT MARKED IMPROVEMENT IS NEEDED IN CYCLE LIFE AND IN CHARGE ACCEPTANCE CHARACTERISTICS
- HYBRID HEAT ENGINE/ELECTRIC FAMILY CAR
  - PARALLEL vs SERIES POWERTRAIN CONFIGURATION
    - PARALLEL HAS 10-15% LOWER EMISSION
  - THE PARALLEL CONFIGURATION IS A SLIGHTLY MORE COMPLEX SYSTEM WITH LESS DESIGN FLEXIBILITY
  - BATTERY RECHARGE EFFICIENCY
    - SMALL EFFECT ON EMISSIONS SHOWN OVER A REALISTIC RANGE OF EFFICIENCY
  - REGENERATIVE BREAKING EFFECTS
    - SLIGHTLY DECREASES DEPTH OF DISCHARGE
    - NEGLIGIBLE EFFECTS ON VEHICLE EMISSIONS
  - TOTAL VEHICLE WEIGHT
    - BATTERY DESIGN GOALS ARE VERY SENSITIVE TO POWER SYSTEM WEIGHT ALLOCATION
  - COLD START EFFECTS
    - FOR HC AND CO INCREASE FROM 15 TO 50% (AVERAGES ABOUT 30%)
    - FOR NO_x SLIGHT DECREASES
- HYBRID HEAT ENGINE/ELECTRIC COMMUTER CAR
  - LOW-WEIGHT REDUCED-PERFORMANCE CAR CAN REDUCE EMISSIONS BY A FACTOR OF 2.5 COMPARED TO THE HYBRID FAMILY CAR
- HYBRID HEAT ENGINE/ELECTRIC BUS AND VAN
  - DIESEL AND TURBINE ENGINES LOOK ATTRACTIVE FOR BUS AND VAN APPLICATIONS WITH MINIMUM DEVELOPMENT EXPENDITURES
  - POTENTIAL IMPROVEMENTS IN HYBRID BUSES AND VANS COULD LEAD TO SIGNIFICANT REDUCTION IN NO_2 AND CO
  - VEHICLE EMISSIONS DATA ARE REQUIRED OVER REALISTIC DRIVING CYCLES ON CURRENT BUSES AND VANS BEFORE COMPARISON OF THE HYBRID MODE CAN BE ASSESSED
  - BATTERY POWER DENSITY AND ENERGY DENSITY SHOULD EASILY BE MET FOR THE HYBRID BUS APPLICATION

Chapter XII

ADVANCED TECHNIQUES IN ELECTRICAL VEHICLES

by

Bohers/Ducrot
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Citroen Automobile Company/Electricity Section
Nanterre, France

Translated for EPA by SCITRAN
(Scientific Translation Service)
Santa Barbara, California, USA

Ladies and gentlemen, my name is Pol Ducrot. I am an engineer with the Research and Development Division of the Citroen Automobile Company — Electricity Section.

Let me, in turn, thank the Philips Company for inviting me here. Thanks to them, I shall report to you on the advanced studies we are carrying out in the field of electric vehicles.

In many fields, from the old drive to the recent SM, Citroen has been in the vanguard — for example, in the area of vehicle-ground communications or oleopneumatic suspension for vehicles of a more modest type.

Thus, it is not surprising that the Company is engaged in significant research in the area of electrical drive particularly.

One of the systems we are studying will be described here.

This study is drawn from a research program carried out in close cooperation with the Compagnie Francaise de Raffinage and the Battelle Institute of Geneva.

As we all know, an electric drive system must include a source of electrical energy, together with a unit to convert it into mechanical energy in a form adapted to the flexibility requirements which characterize drive.

Unfortunately, nobody has yet found an ideal energy source: quiet, nonpolluting, inexpensive, economical to run, very light, and... so simple to design in sketches of future vehicles.

One must look for compromises and optimizations to reduce the often unpleasant disparity between the dream and reality.

This is the purpose of the first part of the presentation, dealing with a cell supplied with air and liquid or gaseous hydrocarbons.

Optimization of the Fuel Cell

This is a solid electrolyte cell, the principle of which is well-known. We know that the working temperature may be selected between 800 and 1000° C, which enables us to avoid using precious catalysts like platinum and to use such inexpensive fuels as hydrocarbons, since conversion becomes quite simple.

The specific performance obtained in this way is quite interesting. The inconvenient aspects of this high temperature should also be mentioned.

The choice of materials is more limited, and the technology is more difficult. The electrolyte thickness should be small, both to reduce the internal voltage drop in the electrolyte and to minimize the mass to be raised to a high temperature, and thus the energy required for the proper temperature.

In parallel with the laboratory tests, an optimization calculation by computer was carried out beginning with the different mathematical models to take into account the effects of the numerous parameters of the cell, which include, for instance:

- arrangement of the converter (exterior or integrated)
- cell dimensions
- maximal and minimal electrolyte temperatures
- activation polarization
- average partial pressure of oxygen in the cathode compartment
- blast-engine output
- heat flux released by parasitic combustion

Photo 1

Each element has a current-voltage characteristic depending on its location in the cell.

Here are the typical characteristics of an element:

Rp . JE designates the voltage drop in the electrolyte related to its ionic conductibility.

ΔEA designates the activation polarizations
ΔEC designates the concentration polarizations

Photo 2

The design shows here the cell with an external converter to be used in models 1 and 2.

Photo 3

Here in Model I, the current drainage occurs perpendicularly to the active faces. We shall term it "cells with transverse current".

Photo 4

In Model II, current drainage occurs parallel to the active faces and the elements are arranged in series by a suitable incorporated arrangement. We shall call it: "cell with internal series arrangement".

Photo 5

Here we see the beginning of Fortran writing of the program relative to the latter model supplied by computer.

Photo 6

Here, by way of example, is one of the curve networks supplied by calculation of Model I with a transverse current. It gives the net output of the cell core as a function of the power-volume ratio for 3 values of the electrolyte thickness. These curves correspond to a cell center which is completely heat-insulated. The preheating of the air is done without loss by heat recovery.

Photo 7

Here is the same network for Model II with internal series arrangement.

Photo 8

Here is a universal optimization curve. The curve in a heavy line is the envelope of the different special curves represented by the dotted lines.

This curve shows the compromise between the power-volume ratio and the net output of the cell core, taking into account the blast-engine.

Electrical Drive System

After this first series of photos, we shall look at a special system of electrical drive by an asynchronous, three-phase motor with casing, supplied by a continuous current source and the result of bench tests carried out in 1968, beginning with a rectangular or sinusoidal modulation drive.

There is no need to emphasize the strength and low price which characterize the asynchronous motor with casing which can be produced in a large range of velocities and outputs.

On the other hand, its adaption to drive starting with a fuel cell requires an "undulator" designed especially for that use.

Photo 9

This is a schematic diagram of the universal control. We can single out:

- the undulator bar control.
- the mechanisms for guiding the bar control. They will receive, regardless of the type of control used, orders for voltage V and frequency fs, and will transform these signals into properly implemented signals in time to obtain the desired voltage amplitude and frequency in the stator. These signals will drive the bar control circuits.

- — operational circuits for supplying V and fs references. These circuits send out the two parameters: the flux and current for the motor.

We then have to select an undulator pattern suited for this application. Two things are certain:

- — It must have "two" return diodes, sometimes called "free wheel diodes."
- — It cannot be of the "series" type where the commutation capacity is placed in series and which does not have the desired operational flexibility.

Even limiting ourselves to extinction by condenser discharge, several extinction circuits can be imagined.

Photo 10

Since some order was needed, they were classified according to the following criteria:

- — the extinction class, termed A or B depending on whether the commutation inductance is or is not traversed by the motor current. This makes the extinction circuit either heavily or slightly dependent on the main current.
- — extinction systems, which may be either direct or alternating.

The table you see illustrates this classification.

At the top, the continuous direct current extinction systems
No. 1: universal. All thyristors are cut off simultaneously.
No. 2 and 3: individual or semi-individual.

At the bottom: "alternating current" extinction systems.

No. 4 and 5: complementary extinction systems — very economical — where lighting one group causes the extinction of the other group.

No. 6 and 7: individual or semi-individual extinction systems, which are much more complex.

Semi-individual alternating current extinction of class B was chosen for our tests, because it had the required flexibility for sinusoidal modulation allowing smaller harmonic content in the current wave.

Photo 11

This picture is taken from our sinusoidal control patent and shows the control ensemble schematically.

To simplify the design, the thyristor protection circuits in $\frac{dV}{dt}$ and $\frac{dI}{dt}$ have not been shown.

We see three control groups U, V and W, with for each:

— T1 and T2 main thyristors
— free wheel diodes D1 and D2
— extinction thyristors S1 and S2
and an inductance/extinction capacity circuit.

Photo 12

Here are the output voltages of the undulator with rectangular modulation. The form factor τ/τ_M is 50% here.

UR - Us shows the composite voltage between phases.

Photo 13

This shows the law governing the change of the cut-off frequency as a function of output frequency.

The cut-off frequency is a whole multiple of 6 fs. When starting, fs = 10 Hz, K = 16, fM = 960 Hz.

We see that K successively takes on the values 16, 8, 4, 2, 1, so that fM never exceeds 2 Khz.

Photo 14

Here we see the theoretical behavior of the current in a main thyristor with sinusoidal modulations at cos $\phi = 1$

Photo 15

This photo is also taken from the same Citroen patent. To obtain the correct cut-off sequence, a special sinusoidal modulation system was perfected.

Here is the principle employed:

Above, diagram A shows a sinusoidal voltage wave 1 with frequency fo.

At instants t1, t2, t3, following periodically at intervals of $\tau_e = 1/fe$, we measure the instantaneous value of the sinusoidal voltage with respect to the reference level 3, a voltage shown by lengths a1, a2, a3...

Thus, a series of asymmetrical triangular signals is generated as shown in diagram B.

We see that their steep fronts b1, b2, b3 occur at instants t1, t2, t3.

- — their heights are proportional to the instantaneous values a1, a2, a3 of sinusoidal voltage 1 measured at these sampling times t1, t2, t3.
- — the oblique fronts c1, c2, c3 all have the same given slope p.
- — in diagram C the triangular signals are transformed into rectangular impulses, each with a width equal to the duration between the rigid front of the corresponding triangular signal and the moment when the oblique front reaches the reference level visible in diagram B.

Thus, impulse I1 has a width equal to the period d1.

So, we see that the impulses I are synchronous with the sampling with frequency fe and are modulated in width in accordance with the instantaneous value of sinusoidal voltage 1 at those times, since the gradient provides a voltage-time linear conversion.

If the reference level is identified with the zero level of the triangular signals, the width d of the impulses is directly proportional to the instantaneous value of sinusoidal voltage measured with respect to level 3.

These impulses I are used to control turning on and off the thyristors inserted in the group commuting an undulator phase.

One of these thyristors is kept on as long as impulse I lasts, and it is turned off by forced extinction during the interval τ_e - d.

Clearly, we need only sample the same sinusoidal voltage 1 at other times offset by $\frac{2\pi}{3}$ to obtain the impulses needed to control two other groups.

A closer examination of the system would show that a second order harmonic would appear with an amplitude proportional to frequency fo; however, a simple and effective correction was introduced by modulating the reference level.

Photo 16

Here we see a simplified diagram of the sampling and amplitude-time conversion apparatus.

- For each impulse f', the sampling condenser C is charged very quickly by the push-button p.
- It is discharged permanently through the constant current source Io.
- The comparator enables us to obtain a logic signal defining the commutation sequence for an undulator phase.

One should note that this signal is the image of the phase output voltage.

Photo 17

Here is the drive curve which shows the behavior of the motor coupling as a function of velocity.

For this motor, a 3.6 Mkgs starting coupling was used for a nominal output of 10 KW with a maximum velocity of 12,000 rpm.

Photo 18

Here is the output measured for rectangular and sinusoidal control.

We see that in the average velocity range the rectangular command system yields a greater output for the laws of cut-off frequency used. This can be explained by the unequal losses in the extinction circuits.

We have also noted increased fatigue in the sinusoidal control extinction circuits — including the extinction condensers — when we attempt to obtain a starting coupling.

For the sake of completeness, I should point out the slightly greater complexity of the sinusoidal control electronics, with little effect on price.

On the other hand, I should stress the coupling modulations derived from pulsating parasitic couplings with rectangular modulation.

In starting, their relative amplitude is on the order of 20%, and the frequency is 60 Hz.

Consequently, we must anticipate a transmission to avoid mechanical fatigue, not to cause resonances and to retain the qualities of comfort and silence so highly valued in electric vehicles.

I shall conclude this presentation with one small detail:

The continuous current source used in these tests was not from a fuel cell.

Thank you, Mr. President, I am prepared to answer questions, to the extent that they are not too compromising — I think you will understand.

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Planche I (1e partie)

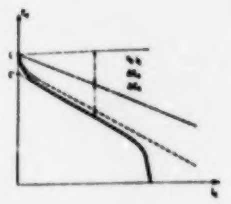


Fig. 16. Electric characteristics of an element.

Ph. 1

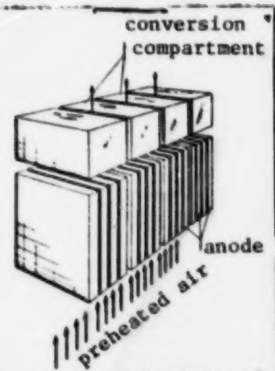


Figure 4. Principle of the heart of a cell with external conversion

Ph. 2

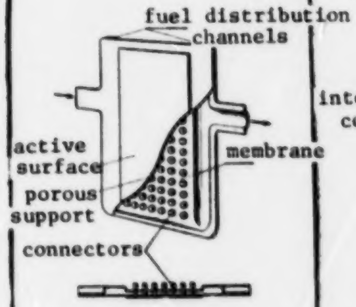


Fig. 1. anode box model I

Ph. 3

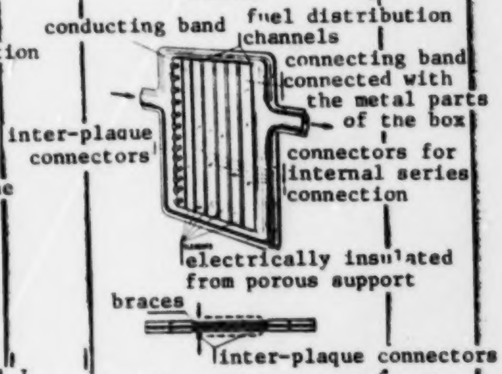
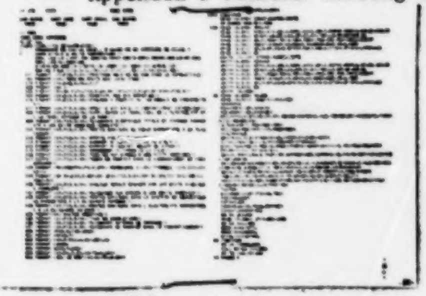

conducting band
fuel distribution channels
connecting band connected with the metal parts of the box
inter-plaque connectors
connectors for internal series connection
electrically insulated from porous support
braces
inter-plaque connectors

Fig. 10. anode box, Module II

Ph. 4

Appendix 5 FORTRAN listing of program II

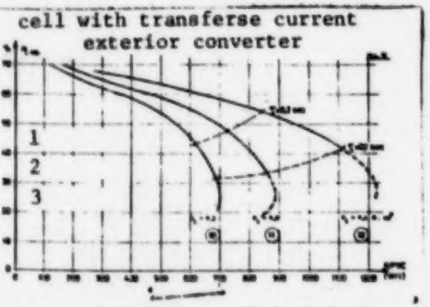


Ph. 5

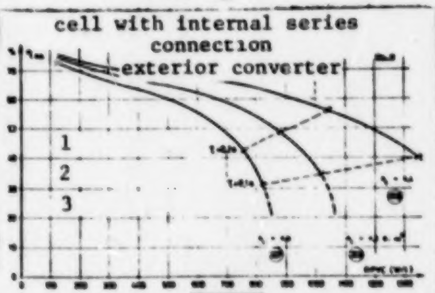
R-8

Legend for Ph. 6 and Ph. 7

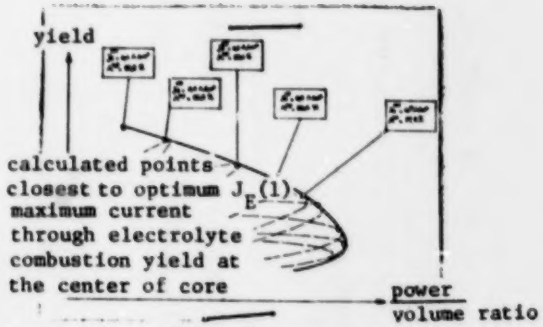
1 — activation polarization: 0.10 V
2 — electrolyte temperature: 800 °C
3 — cathode thickness (In_2O_3): 50u



Ph. 6

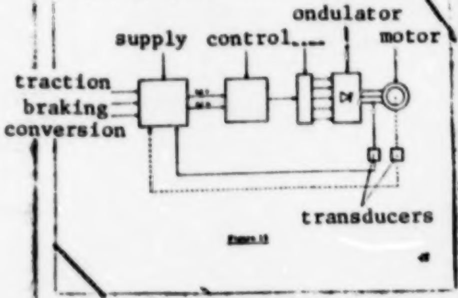


Ph. 7

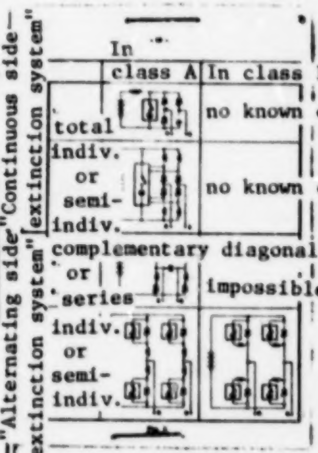


Ph. 8

Planche II (2e partie)



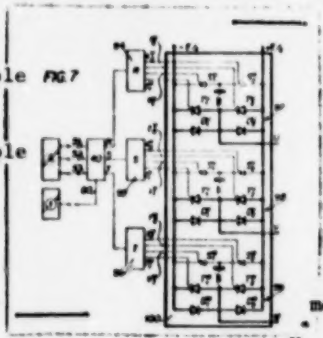
Ph. 9



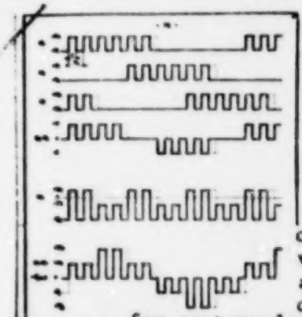
"Alternating side" "extinction system" "Continuous side" "extinction system"

	In class A	In class B
total		no known example
indiv. or semi-indiv.		no known example
complementary diagonal or series		impossible
indiv. or semi-indiv.		

Ph. 10



Ph. 11



output voltages at the ondulator

for rectangular modulation with Fm = 12 Fs and τ/T = 50%

U_R, U_s and U_r voltage delivered by ondulator phases

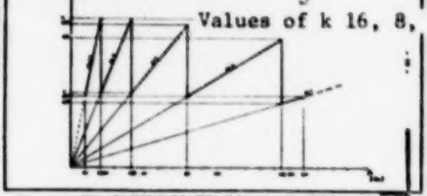
U_R, U_S composite voltage

U_R neutral motor potential

U_R-U_n, U_{phR} neutral-phase voltage of motor

Ph. 1.2

Fig. 2.3 Change in cutoff frequency f_M as a function of output frequency f_S.



Ph. 13

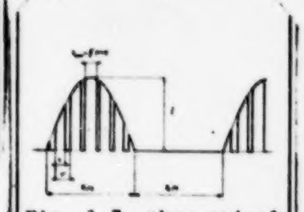
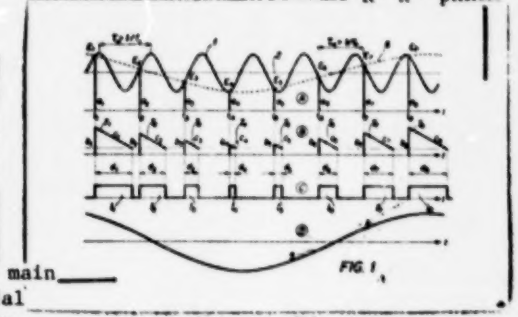


Fig. 1.7. theoretical curve for current in a main thyristor for sinusoidal modulation for a charge with

Ph. 14



Ph. 15

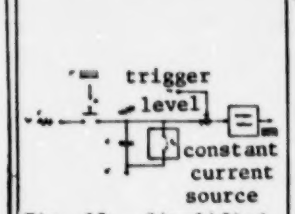


Fig. 12. Simplified diagram of a sampling and amplitude time conversion device

Ph. 16

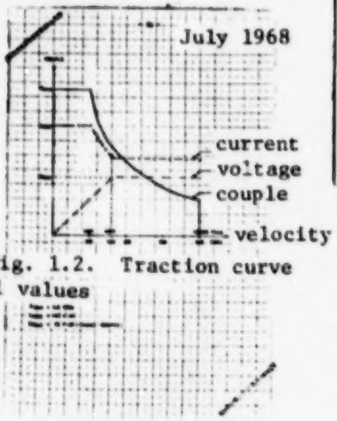


Fig. 1.2. Traction curve

numerical values

Ph. 17

R-8

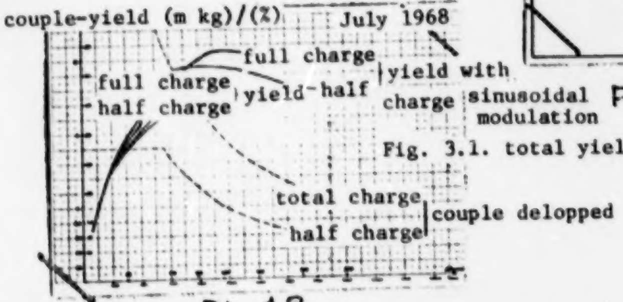


Fig. 3.1. total yields

Ph. 18

Chapter XIII

RESEARCH AND DEVELOPMENT ON A LITHIUM-SULFUR BATTERY

by

Elton J. Cairns,
Section Head, Energy Conversion Section,
Argonne National Laboratories
Argonne, Illinois, USA

Introduction

The Air Pollution Control Office of the Environmental Protection Agency (EPA) has initiated a broad program for the development of low-emissions vehicles. A number of alternatives to the internal combustion engine can be considered to be candidates for use as power plants in such vehicles. Some alternatives are listed in Slide 1 (copies of the slides are attached). The Argonne National Laboratory, under the sponsorship of EPA, is pursuing the development of lithium/sulfur batteries for all-electric vehicles. Some of the goals that ANL has set for itself (in keeping with the EPA goals) are shown in Slide 2. The goal of 220 W-hr/kg cannot be met by any conventional battery, hence the interest in the lithium/sulfur system, which uses a liquid lithium anode, a liquid sulfur cathode, and a molten salt electrolyte containing lithium halides, and operates at 375°C.

Though this program is still in the laboratory stage, some interesting results have been obtained which are worthy of review. The manner in which the lithium/sulfur cell operates during discharge is indicated in Slide 3. It is particularly important to provide for the removal of the product Li_2S from the reaction site, and to supply more sulfur and electrons for reaction, without losing any material to the surroundings. A major portion of our effort is centered around this process.

Experimental Results

An example of a small lithium/sulfur laboratory cell is shown in Slide 4. Voltage-current density and voltage-capacity density curves for such a

SOME ALTERNATIVES TO THE INTERNAL COMBUSTION ENGINE

1. BRAYTON CYCLE (GAS TURBINE)
2. RANKINE CYCLE (STEAM ENGINE)
3. STIRLING CYCLE
4. HYBIRD: HEAT ENGINE PLUS BATTERIES
5. ALL-ELECTRIC: SECONDARY BATTERIES

EPA PROGRAM

HIGH SPECIFIC ENERGY Li/S BATTERY FOR ELECTRIC AUTOMOBILES

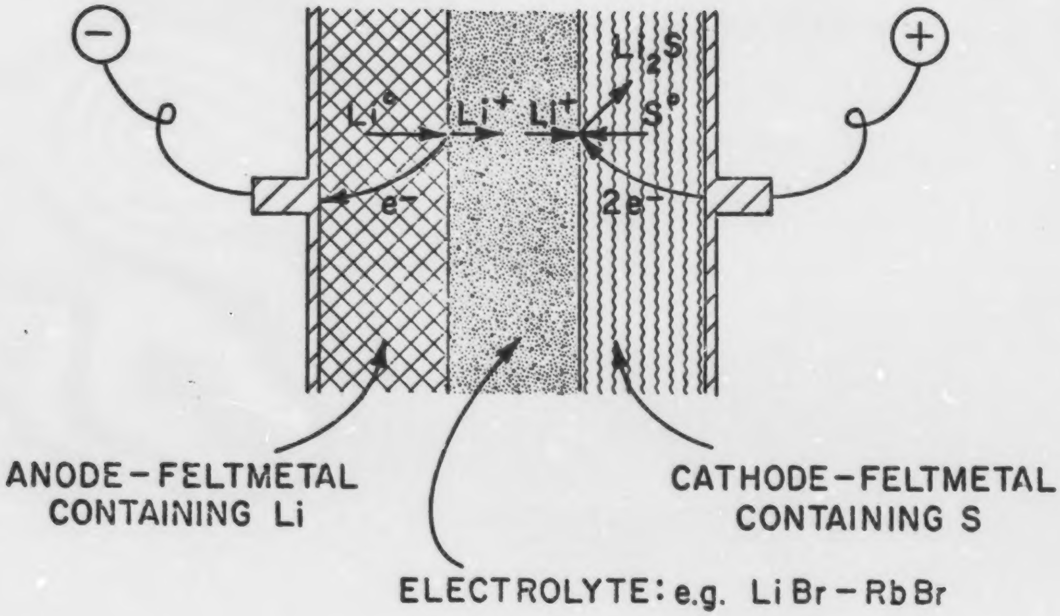
GOALS: 220 W-hr/kg

220 W/kg

1000 CYCLES

$10/kWhr

LOW COST, HIGH SPECIFIC ENERGY



$\ominus$
$\oplus$
Li°
Li^+
Li^+
Li_2S
S°
e^-
$2e^-$
ANODE-FELTMETAL CONTAINING Li
CATHODE-FELTMETAL CONTAINING S
ELECTROLYTE: e.g. LiBr-RbBr

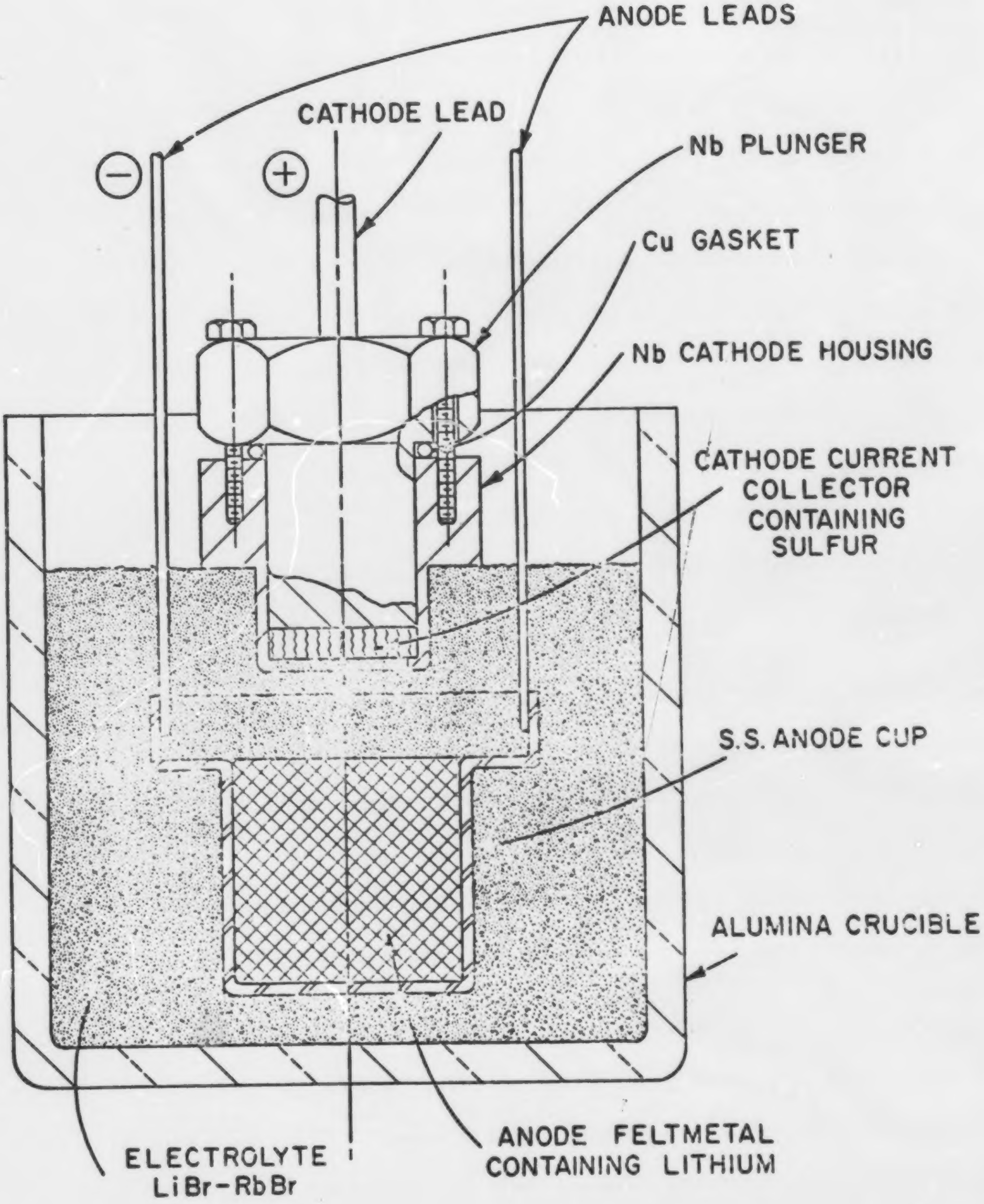
CELL REACTIONS

CELL: $Li/Li^+(ELECTROLYTE)/S(+Li)$

ANODE: $Li^\circ \rightarrow Li^+ + e^-$

CATHODE: $2Li^+ + 2e^- + S^\circ \rightarrow Li_2S$

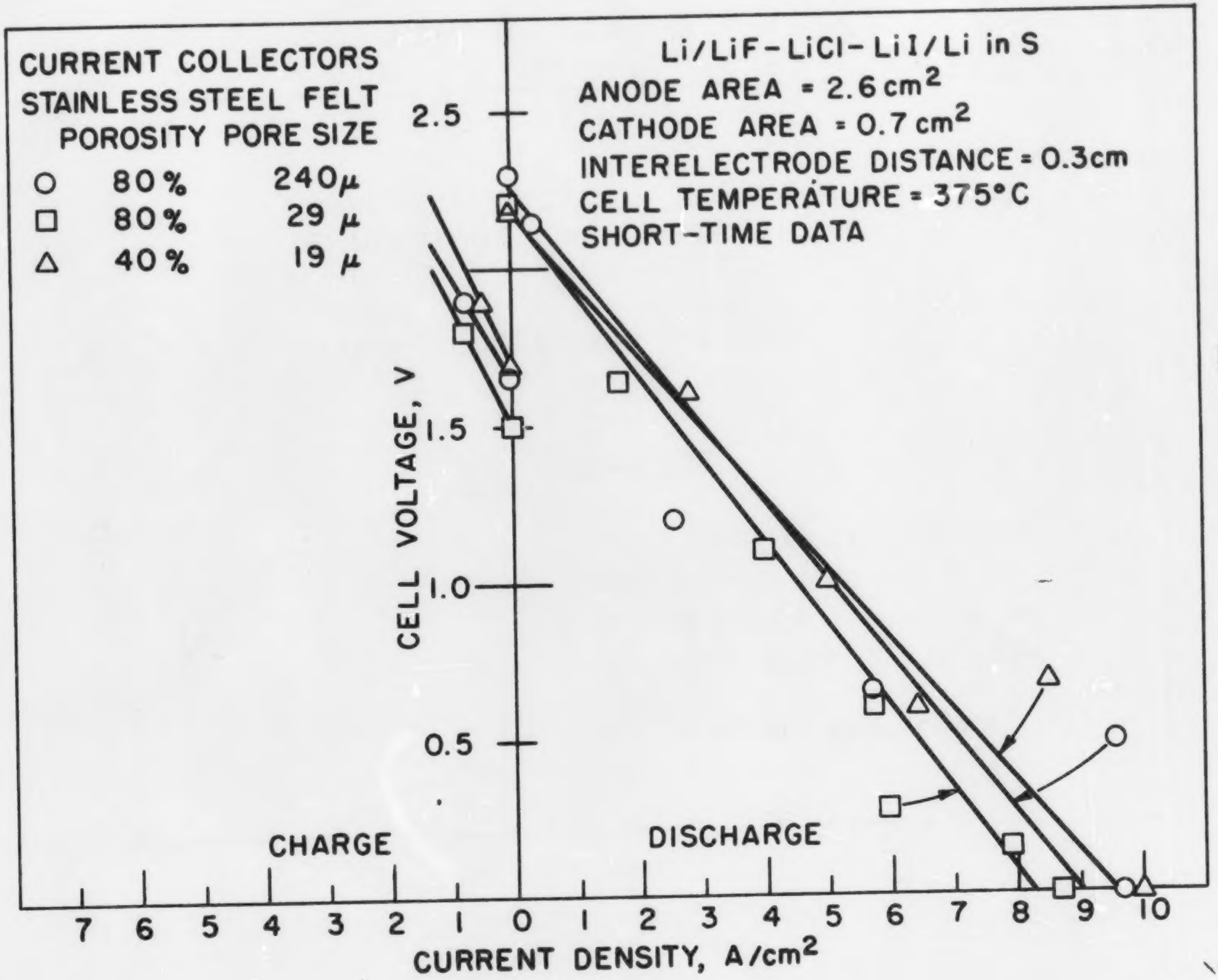
OVERALL: $2Li^\circ + S^\circ \rightarrow Li_2S$

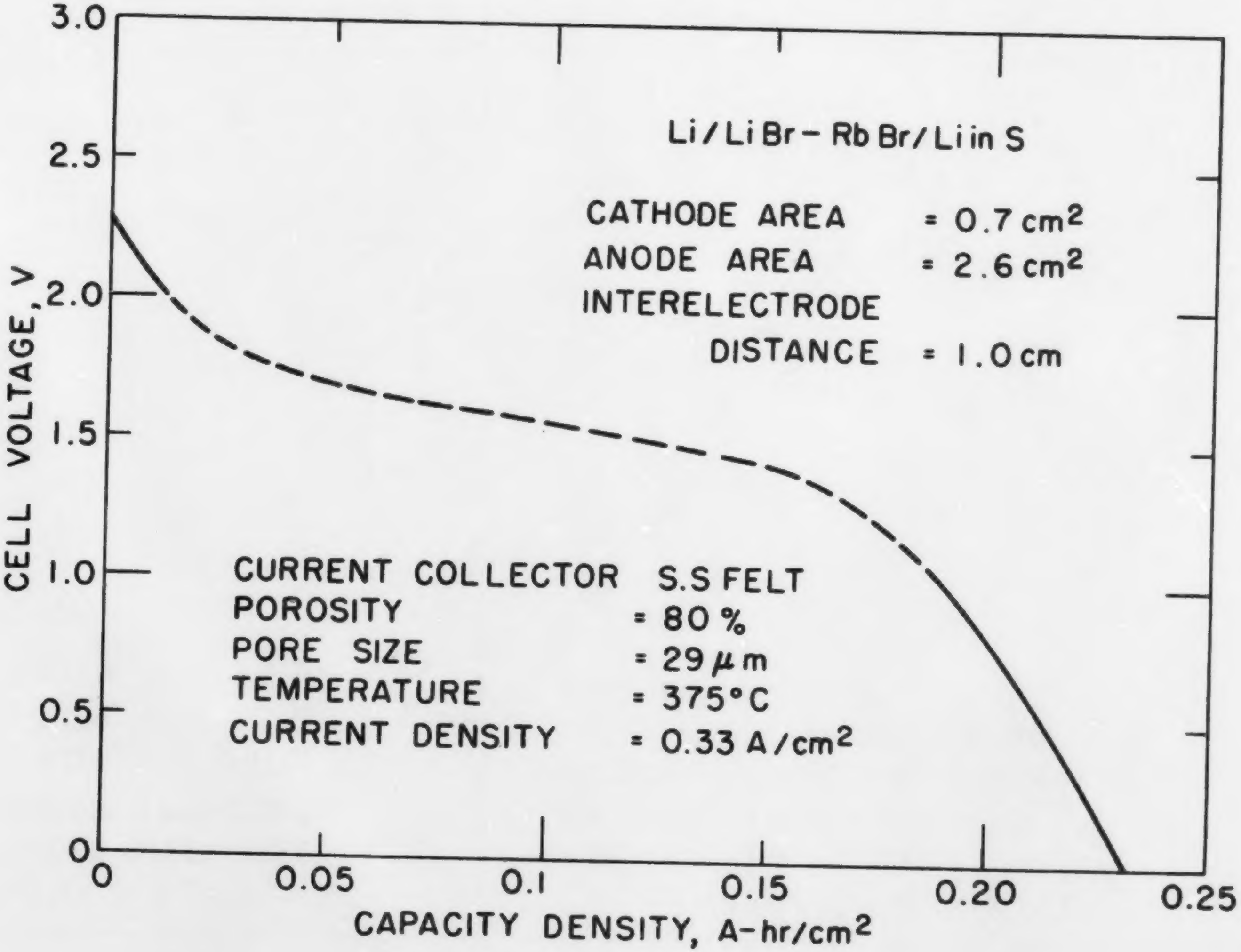

ANODE LEADS
CATHODE LEAD
Nb PLUNGER
Cu GASKET
Nb CATHODE HOUSING
CATHODE CURRENT COLLECTOR CONTAINING SULFUR
S.S. ANODE CUP
ALUMINA CRUCIBLE
ANODE FELTMETAL CONTAINING LITHIUM
ELECTROLYTE LiBr-RbBr
−
+

cell are shown in Slides 5 and 6. The capacity density of Slide 6 is too small for the goals of Slide 2. Therefore, efforts have been made to improve the capacity density by modifying the structure of the cathode current collector, as shown by some examples in Slide 7. The voltage-capacity density curves for constant-current operation of lithium/sulfur cells with a comb, laminated, and enclosed laminated structures are shown in Slides 8, 9, and 10, respectively. The laminated cathode has yielded the highest capacity densities, and the enclosed laminated structure has shown the best cycle life. Another long-lived cathode is the so-called reservoir/cathode, which is comprised of a disk cathode with a space above it filled with sulfur, and an electrolyte-wetted layer of porous material below the disk current collector, to prevent the escape of sulfur.

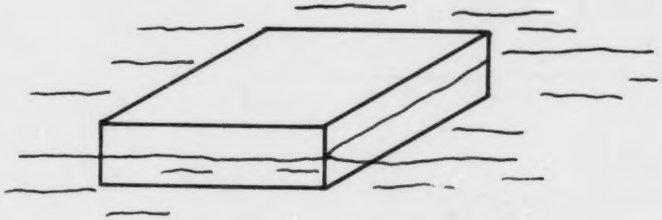
Some of the important performance parameters for various cathodes are summarized in the table of Slide 11. The 220 W-hr/kg goal of Slide 2 corresponds to about 1.2 A-hr/cm^3, or 0.4 A-hr/gm. The cycle lives of the enclosed laminated and reservoir cathodes are within a factor of 2-3 of the goal. It can be seen that some further improvements in the cathodes are necessary before serious scale-up and engineering work can be carried out.

In addition to the experiments with laboratory cells, several other areas of investigation are being pursued, as indicated in Slide 12. The phase equilibrium investigations have as their objective the identification of electrolytes and additives to sulfur which minimize the solubility of sulfur-

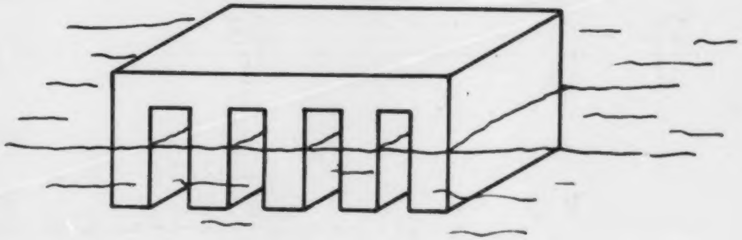

CURRENT COLLECTORS
STAINLESS STEEL FELT
POROSITY PORE SIZE
○ 80% 240μ
□ 80% 29 μ
△ 40% 19 μ
Li/LiF-LiCl-LiI/Li in S
ANODE AREA = 2.6 cm²
CATHODE AREA = 0.7 cm²
INTERELECTRODE DISTANCE = 0.3cm
CELL TEMPERATURE = 375°C
SHORT-TIME DATA
CELL VOLTAGE, V
2.5
1.5
1.0
0.5
CHARGE
DISCHARGE
7 6 5 4 3 2 1 0 1 2 3 4 5 6 7 8 9 10
CURRENT DENSITY, A/cm²



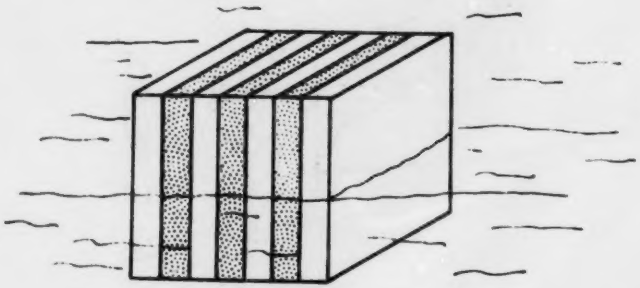
Li/LiBr – RbBr/Li in S
CATHODE AREA = 0.7 cm²
ANODE AREA = 2.6 cm²
INTERELECTRODE DISTANCE = 1.0 cm
CURRENT COLLECTOR S.S FELT
POROSITY = 80 %
PORE SIZE = 29 μm
TEMPERATURE = 375°C
CURRENT DENSITY = 0.33 A/cm²
CELL VOLTAGE, V
3.0
2.5
2.0
1.5
1.0
0.5
0
0
0.05
0.1
0.15
0.20
0.25
CAPACITY DENSITY, A-hr/cm²



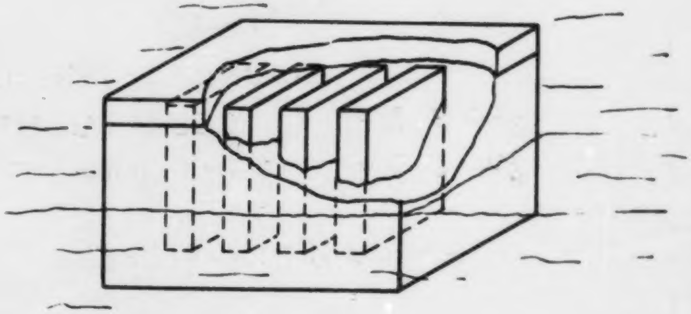
DISK



COMB

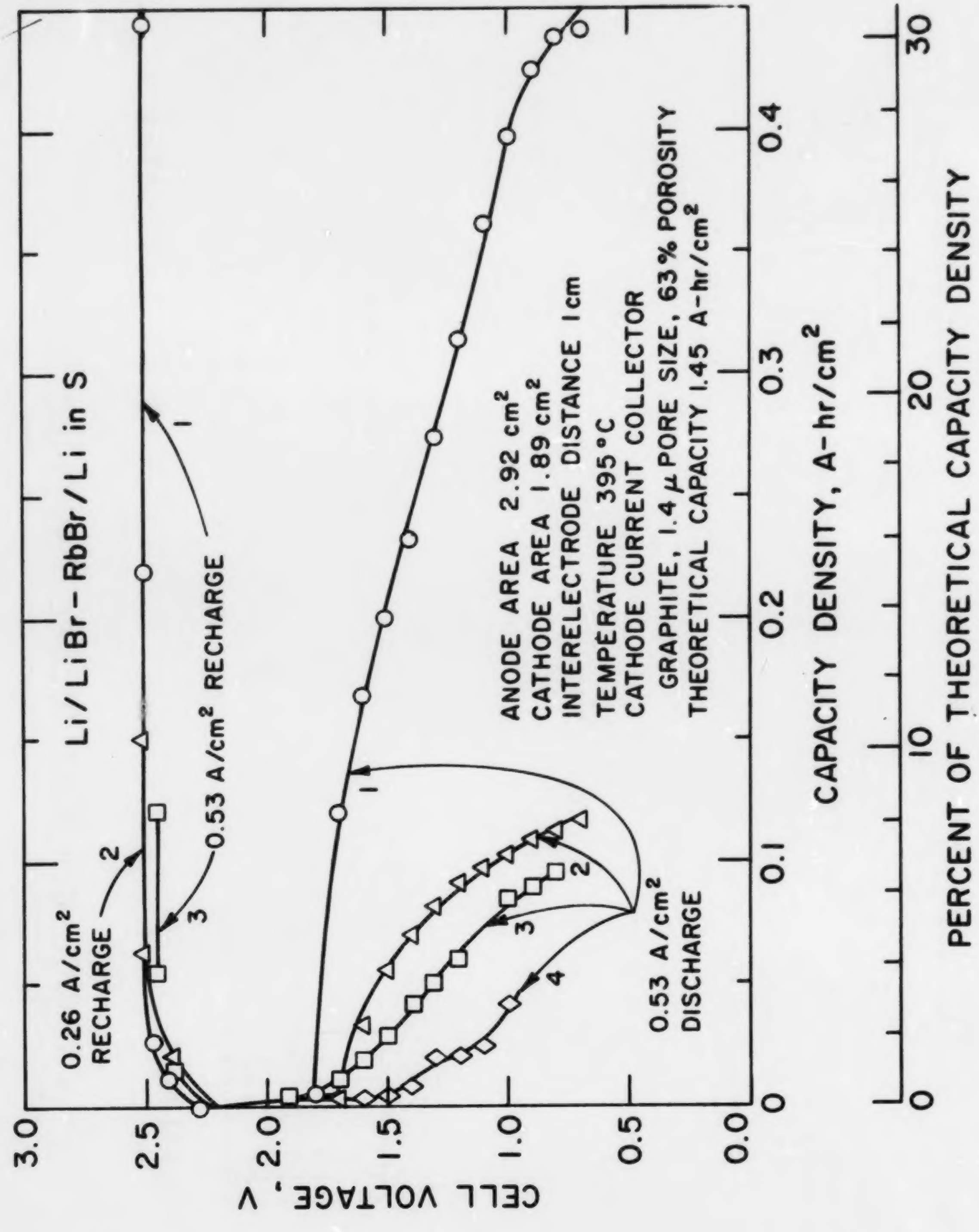


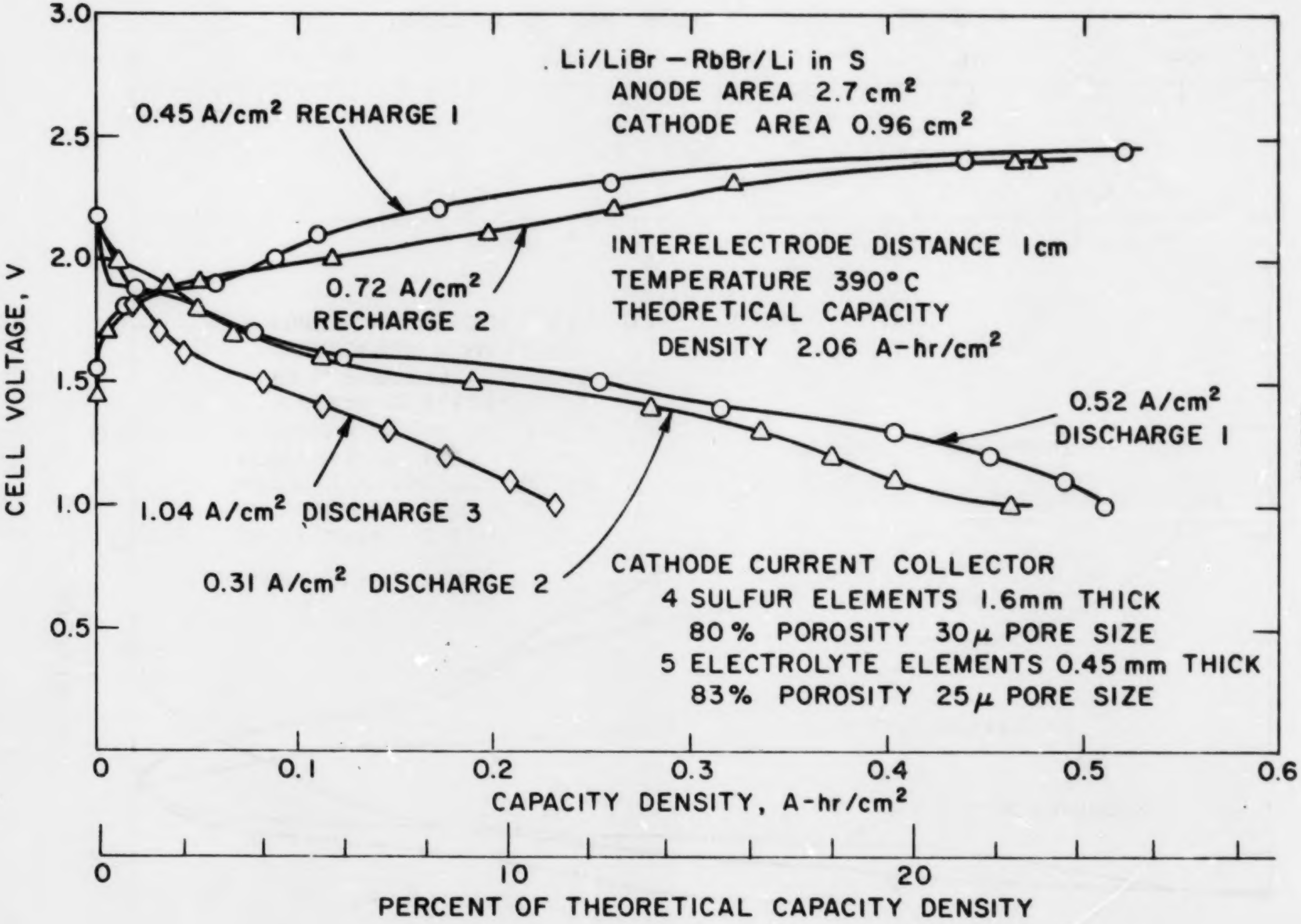
LAMINATED

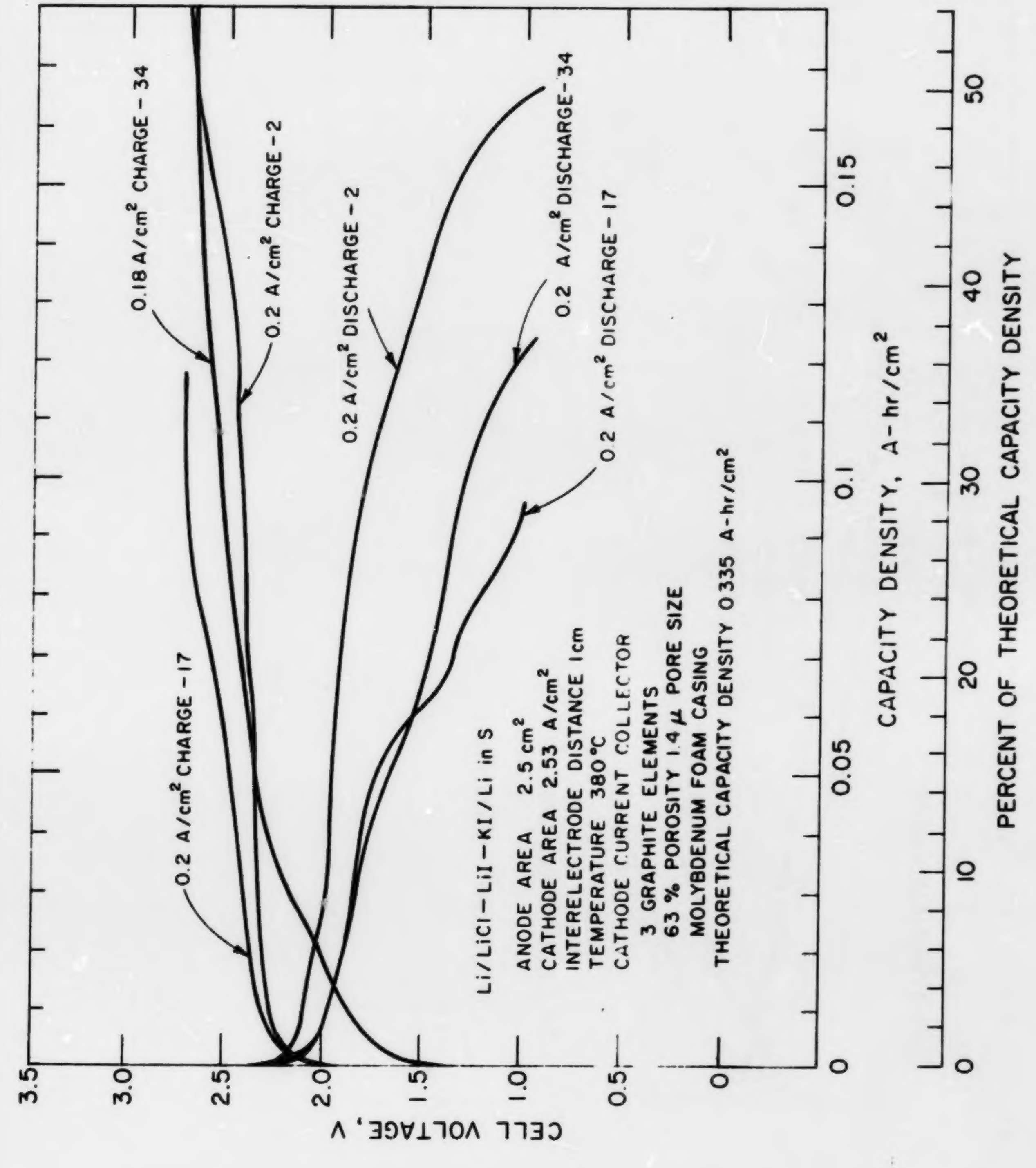


ENCLOSED LAMINATES

CATHODE CURRENT COLLECTOR STRUCTURES


Li / Li Br – RbBr / Li in S
0.26 A/cm^2 RECHARGE
2
3
0.53 A/cm^2 RECHARGE
1
0.53 A/cm^2 DISCHARGE
1
2
3
4
ANODE AREA 2.92 cm^2
CATHODE AREA 1.89 cm^2
INTERELECTRODE DISTANCE 1 cm
TEMPERATURE 395 °C
CATHODE CURRENT COLLECTOR
GRAPHITE, 1.4 μ PORE SIZE, 63 % POROSITY
THEORETICAL CAPACITY 1.45 A-hr/cm^2
CELL VOLTAGE, V
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
0.1
0.2
0.3
0.4
CAPACITY DENSITY, A-hr/cm^2
0
10
20
30
PERCENT OF THEORETICAL CAPACITY DENSITY


3.0
2.5
2.0
1.5
1.0
0.5
CELL VOLTAGE, V
0
0.1
0.2
0.3
0.4
0.5
0.6
CAPACITY DENSITY, A-hr/cm²
0
10
20
PERCENT OF THEORETICAL CAPACITY DENSITY
Li/LiBr — RbBr/Li in S
ANODE AREA 2.7 cm²
CATHODE AREA 0.96 cm²
0.45 A/cm² RECHARGE I
0.72 A/cm² RECHARGE 2
INTERELECTRODE DISTANCE 1cm
TEMPERATURE 390°C
THEORETICAL CAPACITY DENSITY 2.06 A-hr/cm²
0.52 A/cm² DISCHARGE I
1.04 A/cm² DISCHARGE 3
0.31 A/cm² DISCHARGE 2
CATHODE CURRENT COLLECTOR
4 SULFUR ELEMENTS 1.6mm THICK
80% POROSITY 30μ PORE SIZE
5 ELECTROLYTE ELEMENTS 0.45mm THICK
83% POROSITY 25μ PORE SIZE


CELL VOLTAGE, V
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0
CAPACITY DENSITY, A-hr/cm^2
0
0.05
0.1
0.15
PERCENT OF THEORETICAL CAPACITY DENSITY
0
10
20
30
40
50
0.18 A/cm^2 CHARGE - 34
0.2 A/cm^2 CHARGE - 2
0.2 A/cm^2 DISCHARGE - 2
0.2 A/cm^2 DISCHARGE - 34
0.2 A/cm^2 DISCHARGE - 17
0.2 A/cm^2 CHARGE - 17
Li/LiCl-LiI-KI/Li in S
ANODE AREA 2.5 cm^2
CATHODE AREA 2.53 A/cm^2
INTERELECTRODE DISTANCE 1cm
TEMPERATURE 380°C
CATHODE CURRENT COLLECTOR
3 GRAPHITE ELEMENTS
63 % POROSITY 1.4 μ PORE SIZE
MOLYBDENUM FOAM CASING
THEORETICAL CAPACITY DENSITY 0.335 A-hr/cm^2

TYPICAL PERFORMANCE CHARACTERISTICS
FOR VARIOUS CATHODE STRUCTURES

CATHODE	W/cm^{2*}	$A\text{-}hr/cm^{2**}$	$A\text{-}hr/cm^3$	A-hr/gm	CYCLES	LIFETIME hr
DISK	3	0.2	0.6	0.21	<5	<10
COMB	3.5	0.4	0.3	0.15	<2	<5
LAMINATED	4	0.5	0.5	0.17	<10	<20
ENCLOSED LAMINATED	4	0.2	0.15	0.05	419	588
RESERVOIR	2	0.36*	0.15	0.05	>400	>500

*PEAK POWER DENSITY, 1 cm INTERELECTRODE DISTANCE

**ONE-HOUR RATE

OTHER AREAS OF INVESTIGATION

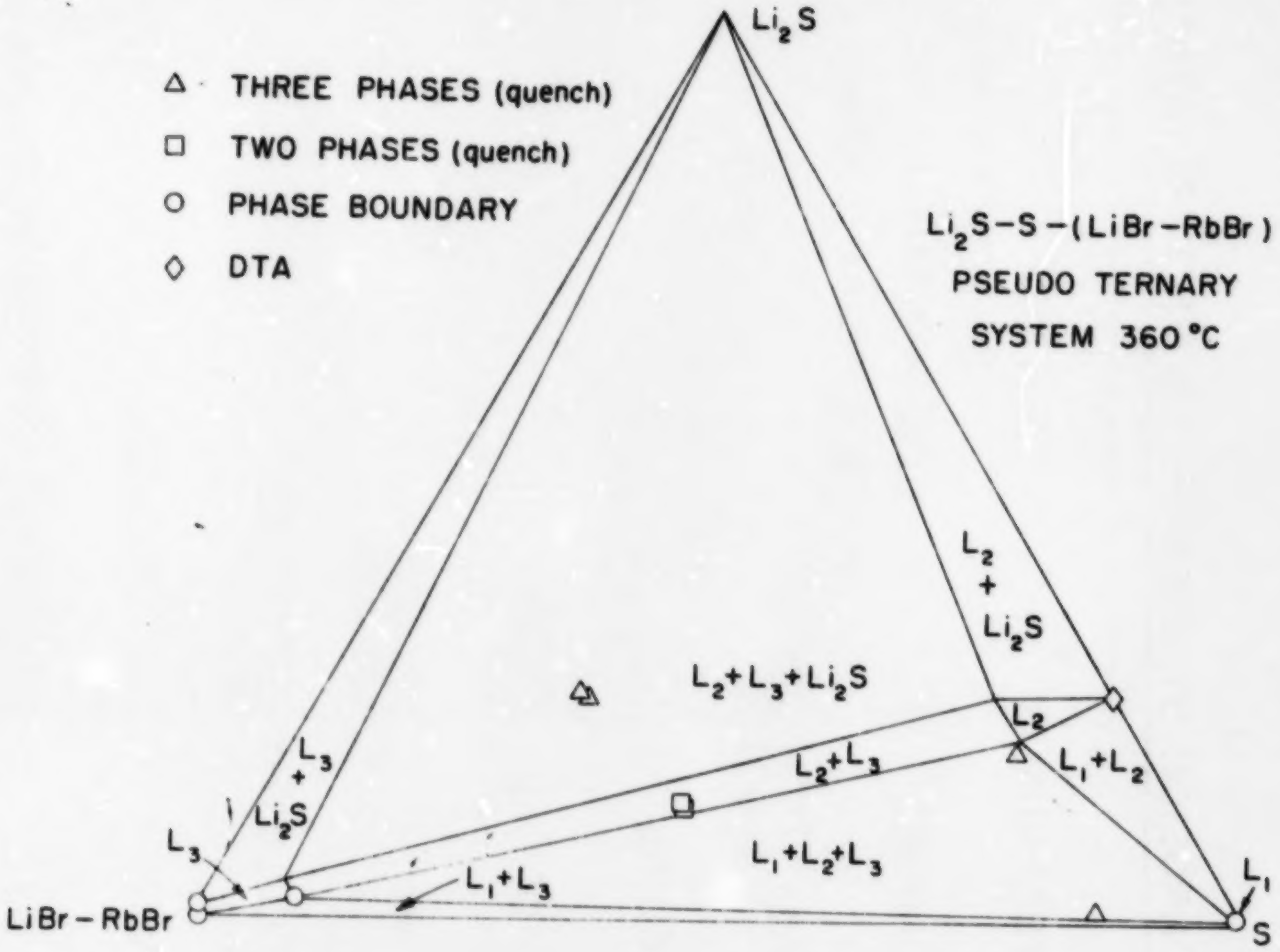
1. PHASE EQUILIBRIA: Li_2S-S-LiX
2. INTERFACIAL PHENOMENA
3. CATHODE MATERIALS STUDIES
4. SOLID ELECTROLYTES
5. MATERIALS EVALUATION

bearing species in the electrolyte. A minimum solubility is desired in order to minimize the rate of loss of sulfur from the cathode. Some typical results are shown in the form of a pseudo-ternary phase diagram in Slide 13. Here, the extent of the phase marked L_3 is to be minimized.

Interfacial phenomena are also important in retaining sulfur. The cathode current collector should be well-wetted by sulfur, but not by electrolyte. The wetting properties of various solids by sulfur and by electrolytes are being studied. Cathode materials (i.e., sulfur, plus various additives such as phosphorus) which have high electronic conductivities, low viscosities, low vapor pressures, and other desirable properties are being sought. Investigations of the suitability of solid lithium-ion conductors for use as electrolytes has recently begun. Materials stability continues to be an important area of investigation. An up-to-date summary of the corrosion rates of a number of candidate materials of construction in lithium-sulfur mixtures and in pure lithium is given by Slides 14 and 15, respectively.

Electric Vehicle Performance Calculations

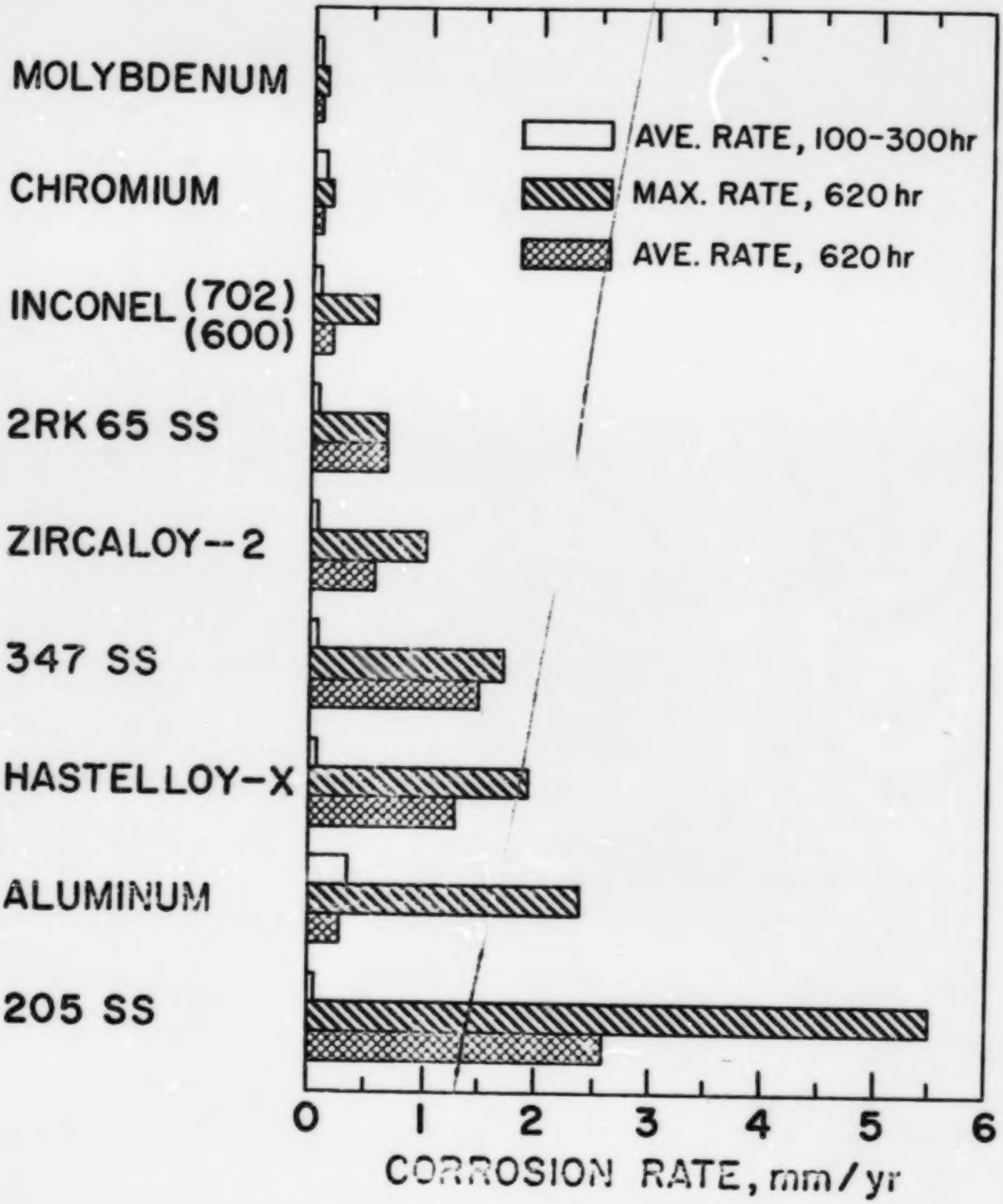
In order to evaluate the potential performance of an all-electric vehicle powered by a lithium/sulfur battery, some computer calculations have been performed for an electric automobile having the characteristics shown in Slide 16. The power requirements for this vehicle were calculated from the equations given in Slide 17, using the values of the constants shown in Slide 18, The driving profiles assumed for the purposes of the



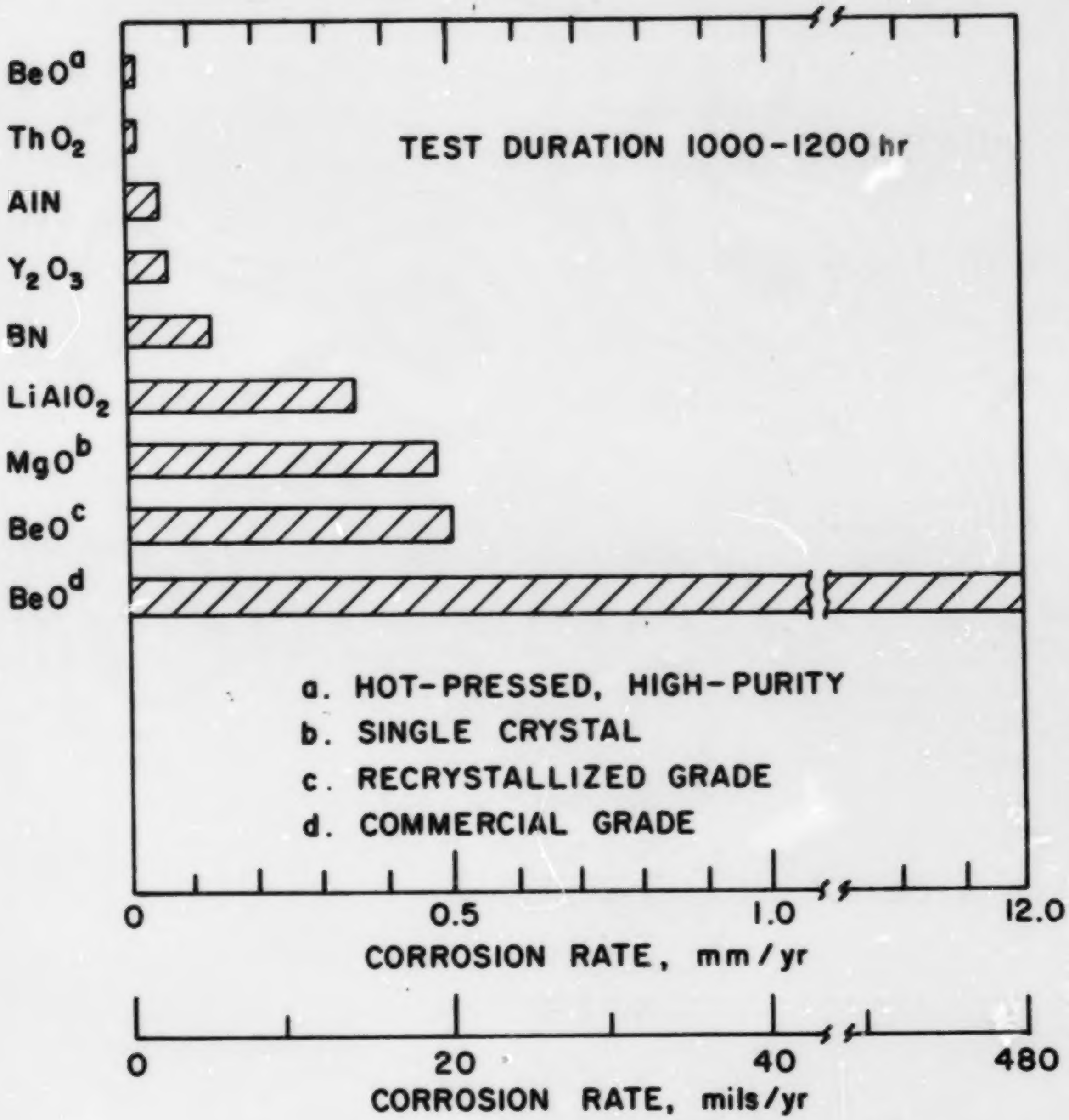
Li_2S−S−(LiBr−RbBr)
PSEUDO TERNARY
SYSTEM 360 °C

XIII-15

CORROSION BY 20 % Li–S MIXTURE 375°C



CORROSION BY MOLTEN LITHIUM AT 375°C



ELECTRIC VEHICLE CHARACTERISTICS

CURB WEIGHT	1588 kg	(3500 lb)
PAYLOAD WEIGHT	227 kg	(500 lb)
BATTERY WEIGHT	397 kg	(875 lb)
SPECIFIC ENERGY (4hr RATE)	220 W-hr/kg	(100 W-hr/lb)
SPECIFIC POWER (1hr RATE)	220 W/kg	(100 W/lb)
ACCESSORY POWER	230 W	
AIR CONDITIONING	4400 W	
POWER STEERING	1900 W	
TOTAL	6530 W	

POWER REQUIREMENTS EQUATIONS

$$P_b = \frac{P_r}{E_m \cdot E_e} + \frac{P_a}{E_{ae}}$$

$$P_r = V(R_r + R_w + R_g + 1.1\ R_a)$$

$$R_r = T_c \times W$$

$$R_w = \rho_A\ C_D\ A_f\ \frac{V^2}{2}$$

$$R_g = W \sin \theta$$

$$R_a = \frac{W}{g}\frac{dV}{dt}$$

VALUES OF CONSTANTS

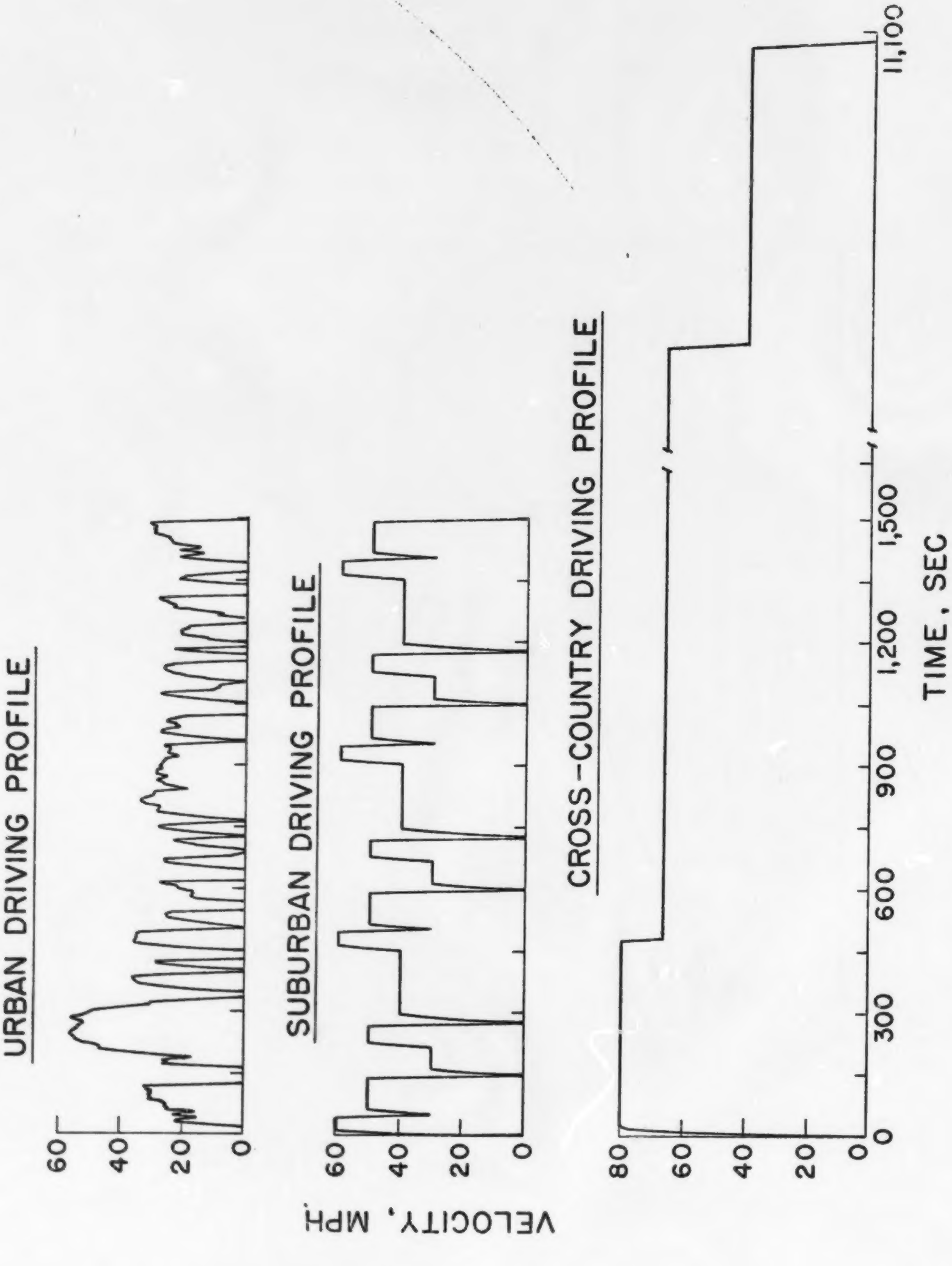
ROLLING RESISTANCE COEFF. (T_c)	0.0175
AIR DRAG COEFFICIENT (C_D)	0.35
FRONTAL AREA (A_f)	$2.32 m^2$
TRANSMISSION EFFICIENCY ($E_m \cdot E_e$)	0.82
TOTAL ACCEL./LINEAR ACCEL.	1.1
AIR DENSITY/g_c	$1.25 \times 10^{-6} \frac{g\ sec^2}{cm^4}$

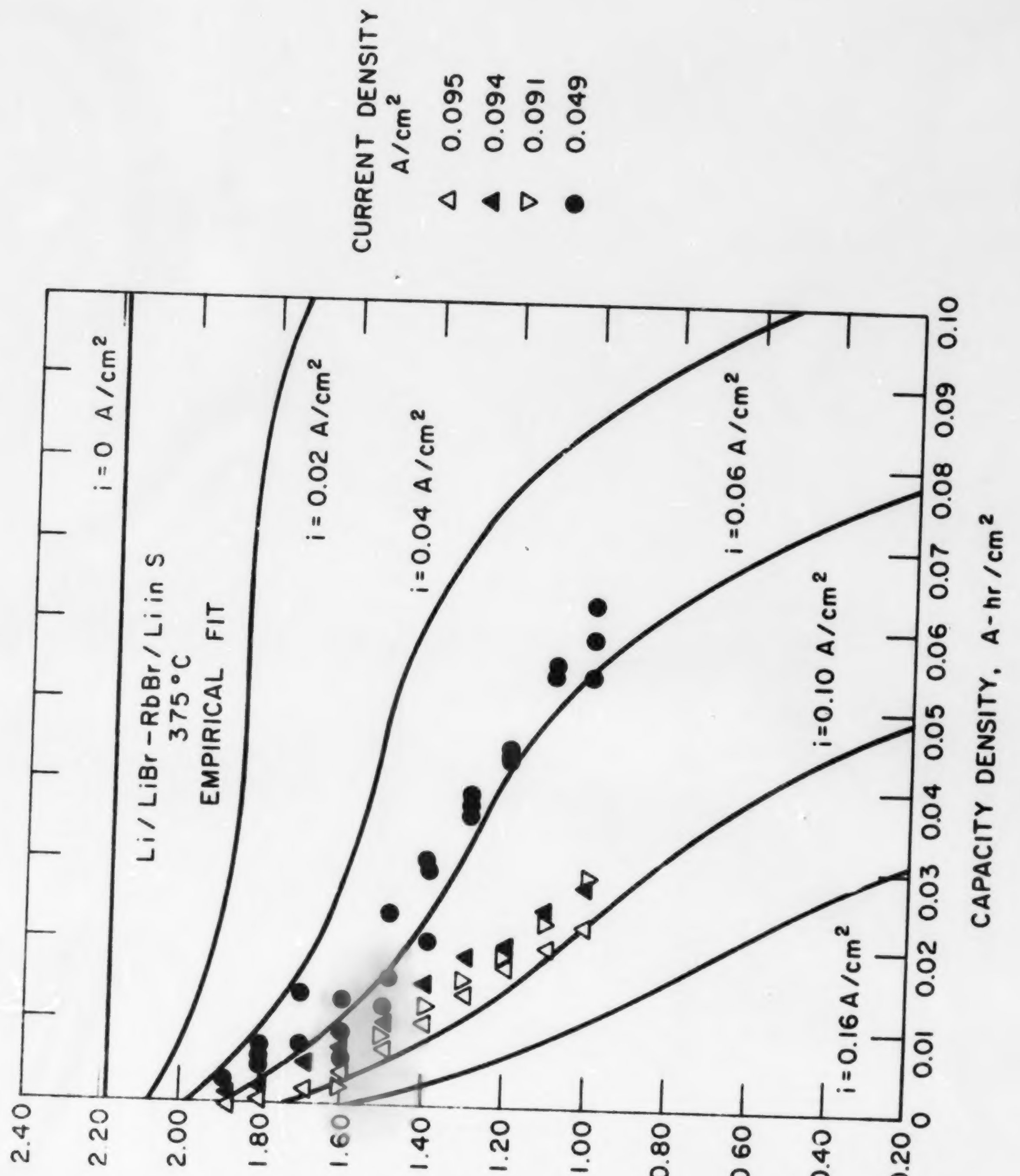
calculations are presented in Slide 19. For each profile the range was calculated for no accessory power for 230 W (lights, heater, etc.) and for 6530 W (air conditioning and power steering added).

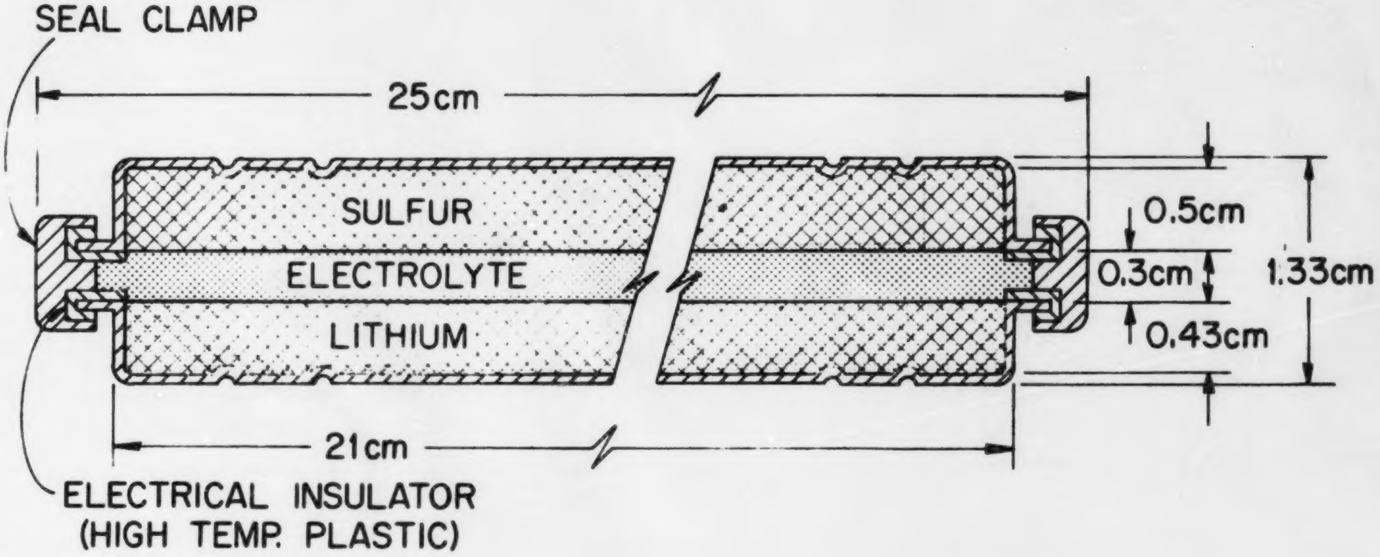
The laboratory results were put into the electric vehicle calculations in the form of E = f (q, i) equations, where E is cell voltage, q is capacity density, and i is current density. Slide 20 shows a set of E - q plots for various i, drawn by the computer. These equations, together with those of Slide 17 and the driving profiles, were combined with a cell and battery design similar to those shown on Slides 21 and 22 to yield the results shown in Slide 23. The ranges shown are less than the EPA goal of 322 km (200 mi); hence, there is a need for improvement in the capacity-density of the laboratory cells and/or the design of the battery. These results do show, however, that electric vehicles based on lithium/sulfur batteries can be expected to have ranges of 150-250 km, if 25% of the curb weight of the vehicle is alotted for batteries.

Acknowledgement

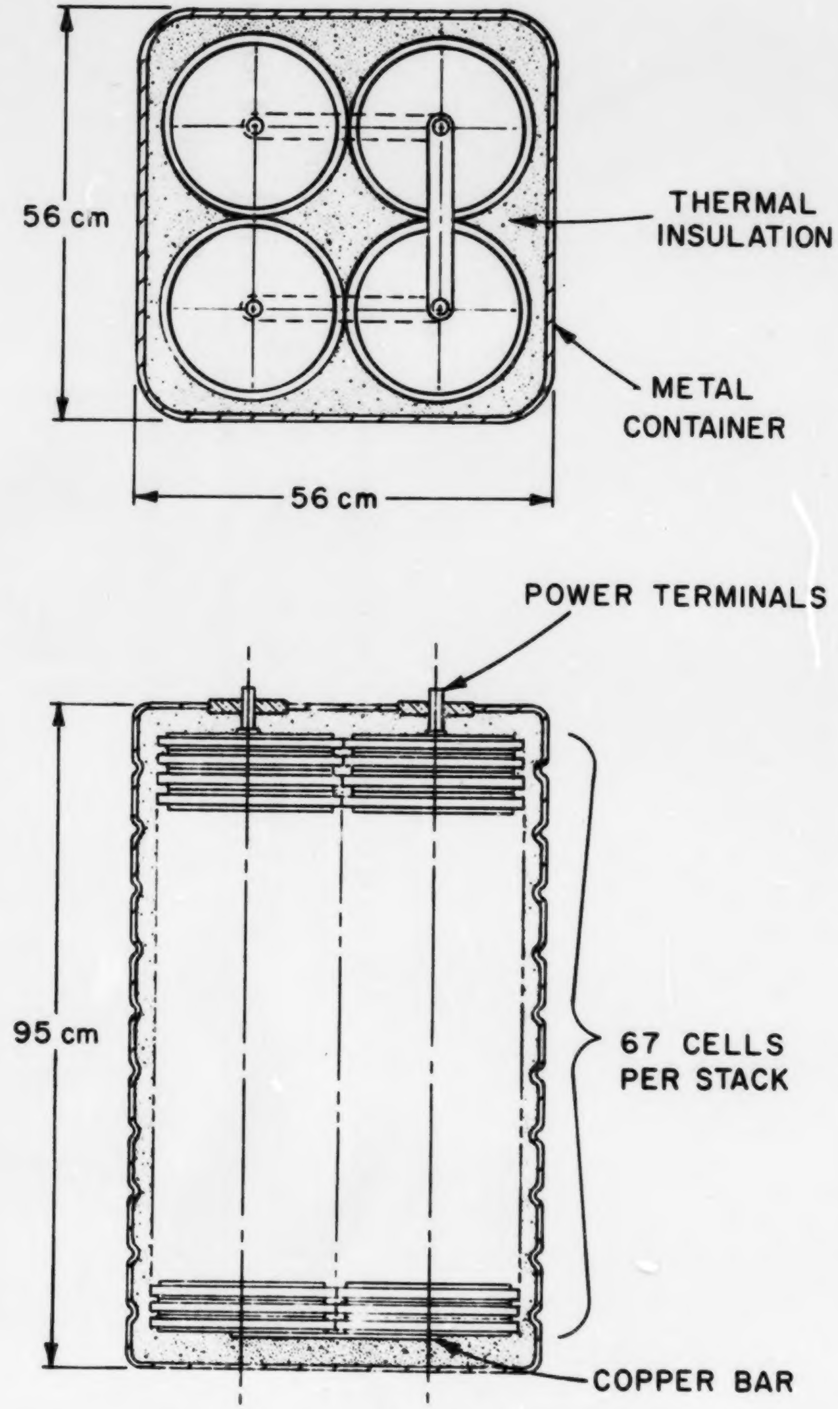
This paper represents a summary of the work of the many people listed in Slide 24. They are the ones to whom credit should go for the accomplishments described above. I also wish to thank the Air Pollution Control Office of the Environmental Protection Agency for support of this program.


URBAN DRIVING PROFILE
SUBURBAN DRIVING PROFILE
CROSS-COUNTRY DRIVING PROFILE
VELOCITY, MPH
60
40
20
0
60
40
20
0
80
60
40
20
0
0
300
600
900
1,200
1,500
11,100
TIME, SEC

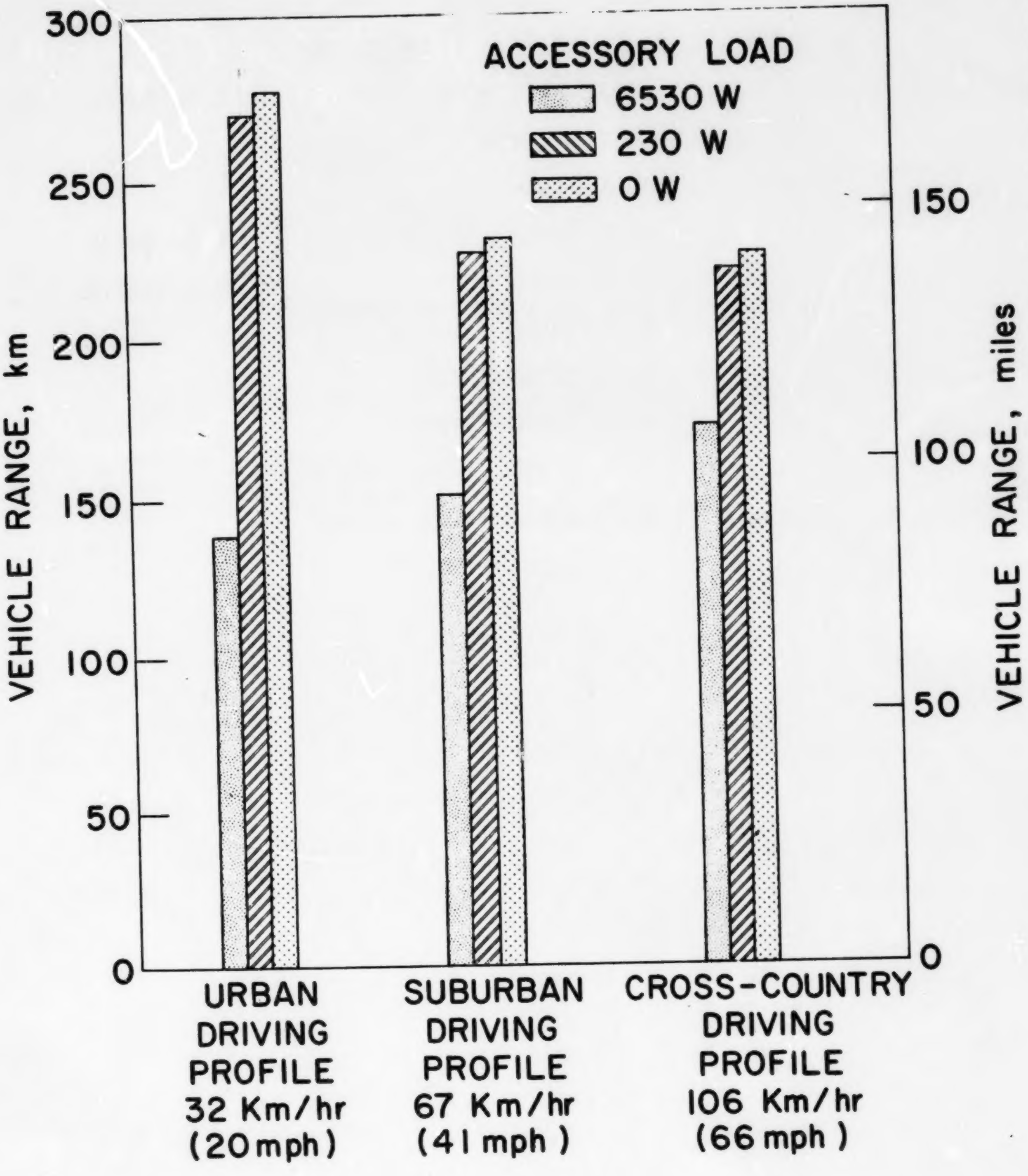

Li / LiBr - RbBr / Li in S
375 °C
EMPIRICAL FIT
i = 0 A/cm^2
i = 0.02 A/cm^2
i = 0.04 A/cm^2
i = 0.06 A/cm^2
i = 0.10 A/cm^2
i = 0.16 A/cm^2
CURRENT DENSITY
A/cm^2
△ 0.095
▲ 0.094
▽ 0.091
● 0.049
CELL VOLTAGE, V
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
0.80
0.60
0.40
0.20
CAPACITY DENSITY, A-hr/cm^2
0
0.01
0.02
0.03
0.04
0.05
0.06
0.07
0.08
0.09
0.10



LITHIUM/SULFUR CELL CONCEPTUAL DESIGN



LITHIUM / SULFUR BATTERY
CONCEPTUAL DESIGN


ACCESSORY LOAD
6530 W
230 W
0 W
VEHICLE RANGE, km
0
50
100
150
200
250
300
VEHICLE RANGE, miles
0
50
100
150
URBAN DRIVING PROFILE 32 Km/hr (20 mph)
SUBURBAN DRIVING PROFILE 67 Km/hr (41 mph)
CROSS-COUNTRY DRIVING PROFILE 106 Km/hr (66 mph)

EPA PROGRAM

R. K. STEUNENBERG

J. P. ACKERMAN

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Chapter XIV

RESEARCH AND DEVELOPMENT PLAN OF ELECTRIC CAR

by

Mr. Shizume
Japanese Automotive Manufacturers' Association
Japan

1. Fundamental Plan

(a) We intend to develop a "new type electric car" for the purpose of use in the town from 1971 to 1975. New type electric cars include the following variations:

(1) Small scale passenger car.
(2) Small scale cargo car.
(3) Medium scale passenger car.
(4) Medium scale cargo car.
(5) Bus

The development cost is estimated about thirteen million dollars. We develop it by concentrating the R & D abilities of national organizations, private automobile companies, universities and so on. Our main purpose to develop it is to protect the environment (air pollution caused by exhaust gas or noise caused by engines).

(b) We will develop it according to the following procedure:

(1) We will develop the experimental electric car by pursuing the optimal structure and performance to accomplish this purpose. We will make the primary experimental electric car and test its performance and safety, etc. for the first three years. Then we will make the complete experimental electric car and not only test its performance and safety but also judge it from the view of technique, social needs or economy for the following two years. The performance data shown later is the performance of primary electric cars which we will develop for the first three years.

(2) We will research and develop several components used to the experimental electric car such as Pb-PbO_2 battery, new type

battery, new type motor (thyristor motor, wheel motor, etc.) controller, plastic body and so on. We will use this result to develop the experimental electric car.

(3) On the other hand we will study the utilization system of electric cars from the view of city traffic system, standardization, optimal total energy system and so on in order to use it well and promote the popularity.

Appendix

National Research and Development Program

(1) Object

From the standpoint of national interest, many fields of industrial technology need urgent research and development which requires a great deal of expense and a long-term period for their fulfillment. It is demanded of the Government to promote such important research and development positively by making plans, bearing research expenses and by organizing research abilities.

In answer to these acute needs the National Research and Development Program System was established in 1966.

(2) Method of Selection

Each project is selected from the projects proposed by governmental organizations, national laboratories or private companies under the following criteria.

(a) The project is very important and immediately needed to achieve leveling up of industrial construction, effective exploitation of natural resources or preventing public nuisance.
(b) Its techniques are advancing and have effective spin-offs.
(c) Large amount of financial resources over a long period are needed to research and develop the techniques and, these, in addition, are subject to heavy risks.
(d) The target of R & D of the techniques must be set and also the technical method to achieve it must be forecasted.
(e) The R & D of the techniques requires the concentration of the R & D abilities of national organizations, private companies, universities and so on.

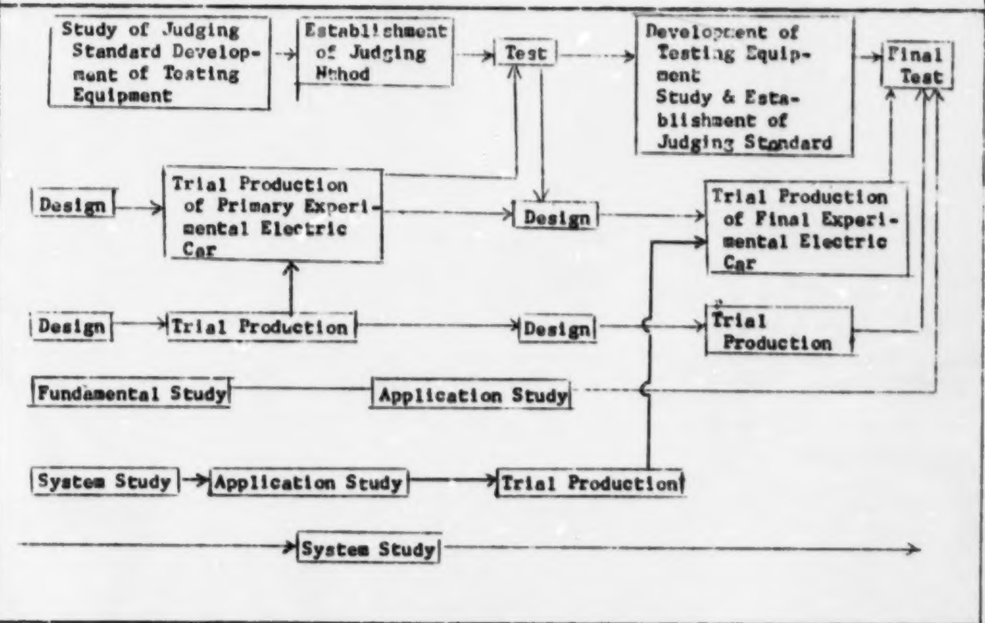
Now the National R & D Program System (NRDPS--known as Ogata-project--in Japanese) includes the R & D of "Magneto-hydrodynamics generator," "Large-scale digital computer," "Desulfurization process," "New process for olefin production," "Sea water desalting and by-product recover," and "Remotely controlled undersea oil drilling rig." From April 1971, "Electric car," "Jet engine used to civil aircraft," and "Pattern recognition system" will be starting to develop.

(3) Method of Management--A special organization for managing the System has been established in The Agency of Industrial Science and Technology (AIST). The organization manages each of six projects. The following five headings has the responsibility of the management.

(a) Selection of R & D project.
(b) Planning the actual programme.
(c) Selection of sub-contractors.
(d) Operation of the System.
(e) Management of the achievements.

Another organization, such as the advisory committee or evaluating committee, assists the organization in managing the System.

2. Research and Development Plan of Electric Car



	1971	1972	1973	1974	1975	Remarks
Establishment of Judging Standard & judgement of Electric Car	Study of Judging Standard Development of Testing Equipment	Establishment of Judging Method	Test	Development of Testing Equipment Study & Establishment of Judging Standard	Final Test	National Laboratory
Development of Experimental Electric Car	Design	Trial Production of Primary Experimental Electric Car	Design	Trial Production of Final Experimental Electric Car		Car Maker
Development of Several Compoents	Design	Trial Production	Design	Trial Production		Private Company & National Laboratory
	Fundamental Study		Application Study			
Decelopment of Charging System	System Study	Application Study	Trial Production			Private Company
Study of Utilization System		System Study				National Organization
Development Cost (Million Dollars)	1.25	3.64	3.97	2.56	2.0	13.42 (TOTAL)

3. Specification & Performance of the Experimental Electric Car

(The End of 1973)

	Cargo Car		Passenger Car		Bus
	Small Scale	Medium Scale	Small Scale	Medium Scale	Large Scale
Passenger + Payload (kg.)	2 + 200	2 + 1,000	4 or 2 + 100	5 or 3 + 300	60 - 80 persons
Total Weight (kg.)	1,100	3,500	1,000	2,000	15,000
Maxmum Speed (km/h)	70	70	80	80	60
Range (km)	130 - 150	180 - 200	130 - 150	180 - 200	230 - 250
Acceleration Ability (0 - 30 km/h) (Sec)	5	5	4	3	8
Climbing Ability (Speed of 6 degree slope) (km/h)	40	40	40	40	40

Chapter XV

STUDIES BY FIAT ON THE ELECTRICALLY-DRIVEN AUTOMOBILE

by

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Electrically-driven automobiles have a long history at Fiat. As early as the last world war, some electric cars were built on chassis and bodies of that period.

Within the framework of research aimed at reducing air polution, in 1961 Fiat began new studies on the possibility of using electrical drive in the light of current technological possibilities.

For the first experiments, we used vehicles provided with a clutch and gearbox, namely: a Fiat 200 van and a Fiat 1100 automobile.

In orienting these studies, we took into consideration the various possible drive systems with an eye to defining the best applications for use in automobiles, especially urban areas. We found that such systems must satisfy the following conditions:

1. High overall output of the whole drive system, i.e., of the motor and its control system under all vehicle operating conditions, especially in stop-and-go city driving.

2. Capability of quick response in traffic. Thus, the following were deemed necessary:

- fast acceleration;
- automatic braking effect when accelerator is released (similar to effect in thermal motor) with possibility of later energy recovery.
- possibility of velocity control and coupling with no discontinuity or with a discontinuity contained within limits judged acceptable for thermal motor vehicles.

3. Availability of high specific outputs even at small rpm rates to allow good motor operation.

This characteristic enables us to bring into play certain laws of motion using appropriate control to reduce acceleration time, taking into account passenger comfort.

These conditions must be accompanied by feasibility of each element in the system, low price, and inexpensive maintenance.

In the drive system of the vehicle mentioned above,we used a compound excitation motor where one of the coils can be supplied independently. The armature has no dissipation element: velocity regulation depends only on variation of the current in the independently-supplied coil.

This system results in variable motor velocities in a continuous fashion over a range of values which, through special design features, has proven sufficiently broad. Moreover, this system has, naturally, the possibility of dynamic braking by energy recovery with a braking effect similar to that of a piston engine. In fact, under over-excitation conditions with respect to the velocity imposed by the car, the motor shifts to operating as a generator with no discontinuity.

The energy expended for regulation is practically negligible and the output of the drive system, identified with that of the motor, assumes values close to 95% under normal operating conditions.

The output in the transitory regime may reach the same values by similarly setting the de-excitation gradient in correlation with velocity.

In this system, the motor velocity must not drop below a certain regime corresponding to maximum excitation; consequently, the installation must be completed with a gearbox and transmission. Nonetheless, the system has the advantages of durability, economy and high overall output, which were the bases of the research orientation.

We next attempted to simplify the operation of experimental cars by eliminating the transmission and gearbox.

For that purpose, we added a nondissipating regulation system to the armature circuit to permit the motor regime to vary from zero velocity to the velocity when regulation of the independent field begins.

A system of this type was used on a car derived from the Fiat 850 model.

The characteristics of the car are as follows:

— Car	Fiat 850 (with structural modifications)

— Total unloaded weight	1025 kg
— Batteries	lead, special type with plastic vat
— Battery weight	320 kg
— Battery voltage	96 V
— Motor	Fiat, 6 poles, compound excitation
— Nominal output peak	21 CV
— Maximum peak output	45 CV
— Maximum velocity	72 km/h
— Acceleration	from 0 to 50 km/h in 8 seconds
— Range at 60 km/h on level road	65 km

In order to ascertain more realistically the performance of the regulation system developed for a small light car especially suited for use in town, a Fiat 500 car was transformed to electrical drive.

To simulate the performance which could be supplied by these future, high energy density, light-weight batteries, the car was fitted with a limited number of lead batteries.

The characteristics of the car are as follows:

— Car	Fiat 500 (with structural modifications)
— Total unloaded weight	730 kg
— Battery weight	160 kg
— Battery voltage	96 V
— Motor	Fiat, 6 poles, compound excitation
— Nominal output	21 CV

— Maximum peak output	45 CV
— Maximum velocity	80 km/h

The operational characteristics of this car, aside from its range which was not significant in this study, were judged satisfactory. The car was easy to drive in city traffic, due to the outstanding pick-up.

We should point out that these cars were produced especially for the purpose of testing certain drive systems. These systems, moreover, allow us simply to increase the performance to the point of making them similar to those of traditional vehicles for use not only in the city.

We are at present producing a vehicle fitted with a drive system with an asynchronous motor fed by batteries through a static convertor (produced in collaboration with Philips Co. of Milan).

This experience was regarded as a prelude to the use of such a system in buses.

In particular, we plan the production of hybrid buses with batteries recharged by an electric generating group, which could be brought about by a gas turbine.

The drive systems tested so far use motors of almost entirely conventional types, even if they are designed with particular features to make best use of certain characteristics.

The relative control modes consequently conform to the need to adapt the characteristics of the motor to the requirements of driving the vehicle — elements which can not always be reconciled.

At this point, it was regarded as opportune to approach the problem of a drive system as a whole anew and to revise the motor design by bringing it closer to the requirements of a vehicle through more rational regulation, i.e., considering the structure of the motor as intimately connected to the system of control.

Current Fiat research programs on electrical vehicles are based on these principles.

Chapter XVI

ELECTRICAL VEHICLES WITH FUEL CELLS: WHY AND HOW?

by

J. Beslier
Chief of Electrical Equipment Research
Peugeot Automobile Company
Paris, France

Chapter XVI

ELECTRICAL VEHICLES WITH FUEL CELLS: WHY AND HOW?

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In order not to impair road traffic, the electric vehicle should have a performance close to that of other vehicles on the road. It must have sufficient speed, good acceleration ability, and a normal action radius. But strengthening the car structure to ensure safety in case of impact makes the chassis heavier and will make the construction of small vehicles more and more difficult. All this leads to a vehicle having medium-sized dimensions, equipped with a powerful generator, with large energy capacity. Only the fuel cell meets the requirements of the problem.

The associated Companies PEUGEOT-ALSTHOM have signed an important contract for research with ESSO-USA, which should lead to a methanol-air cell that would be suitable for the contemplated vehicle.

In an electric vehicle, the energy consumed by the motor comes from a generator which is to the electric motor what the boiler is to the steam engine. To answer the problem posed, which is the reduction of pollution, one needs a nonpolluting generator and a silent motor transmission unit.

What Vehicle to Make?

Before examining each of these points, we must define the main characteristics of the desired vehicle in order to select the best adapted elements. In fact, we may attempt to produce a vehicle answering a very special list of specifications (delivery vehicle, taxi, vehicle for city use, small public transportation vehicle) or, on the other hand, we may try to convert the standard vehicles presently used which are included in the various possible categories. We shall not deal here with heavy weights and other large specialized vehicles which are not customarily produced by the Peugeot Company.

Very diverse considerations will influence the choice. They deal with industrial requirements, technical capacities, market demand and the norms imposed by regulations.

Effects of Regulations

As far as the latter are concerned, it is undeniable that they play a large role in the work of the Research Section. Whether it is a question of passive or active safety, nuisance caused by the vehicle or its operational features, all these problems have caused significant chassis changes both with regard to structure and equipment.

Improved safety in case of collision involves strengthening the structure. To withstand front, back, and side collisions, while providing survival space, there must be more and more space for the structural elements which are indispensable for protection of the main body. The latter will tend to increase to permit the installation of protective devices for the passengers. Another consequence of this strengthening is a substantial increase in chassis weight.

All this leads us to believe that it will become increasingly difficult to reduce vehicle dimensions.

Market Demand

It is undeniable that the vehicle best corresponding to the demands of European consumers is the average-sized vehicle. Very small cars have had only ephemeral or limited success. They remain relatively expensive, especially when the price is compared with their very limited space, performance, and comfort capacities. By a very small vehicle, we mean vehicles designed to carry two persons, with a minimum of exterior accoutrements.

The average-sized vehicle owes its success to the fact that, for a price many can afford, it offers a good compromise allowing practical and comfortable use in diverse business or family circumstances.

This success does not seem to be diminishing, and it leads us to think quite naturally that a new vehicle will have a greater chance of market penetration if it corresponds to the needs of the consumer — that is, if it is in the average-sized bracket.

Industrial Requirements

As far as industrial installations are concerned, it is apparent that the replacement of the internal combustion motor by an electric generator-motor unit will involve very complex and costly reconversion problems.

For these reasons, it appears inevitable that the classical vehicle will continue to be produced in parallel. The new investments required will be very high, and traditional equipment will have to be written off.

The last factor coming into play will be purely technical in nature: what motor, and especially what generator, will be used?

The Motor

Many studies have been made around the world on developing electrical motors suited for automobile propulsion. This is a very special problem, because the conditions of use are quite different from those for industrial motors and locomotive engines. For the latter, the weight and dimensions are not as important as in the automobile, since longevity must be considerable: 2000,000 km represents the yearly use of a locomotive. On the other hand, the automobile motor must be able to be mass-produced and must be light and compact. Finally, it must be suitable for the various operating regimes encountered by a vehicle used in different circumstances and climates, by demanding or careless customers.

Motors can be classed into two large categories, according to whether they operate on direct or alternating current. The development of semiconductors enables us to improve flexibility and transmission output and to contemplate the production of DC motors with no collector or supplying AC motors from a DC source. Finally, it is possible (and may be of safety interest) to use the motor for braking. Finally, let us note that the motor output can be divided to activate two motorized axles or four motorized wheels.

The subject of motors is quite vast. Much research remains to be done before we can draw up a complete, technical, and economical balance sheet for the unit comprised of the motor and the electronic power circuits.

The Generator

The sources of electrical energy, or generators, fall into two categories: storage batteries or secondary generators which only restore the energy which they store in electrochemical form and cells or primary generators, which produce electrical energy directly from fuel.

Storage batteries are well known, especially the lead storage battery whose low mass-energy can be improved only slightly. Many other types of storage batteries have been studied or are being tested. It appears that real progress can be made, but at the cost of hitherto undetermined mechanisms. Improvements are also possible in the area of recharging, the duration of which is a significant handicap. The latter is still encumbered by the inevitable need for a complicated infrastructure connecting the users with a distribution network, using a counter for tabulating the energy absorbed.

Cells are also the object of many studies, and various fuels have been investigated. Aside from dangerous, difficult to handle products, or costly products, there are a number of possibilities. The most interesting may be methanol combined with atmospheric air: it is a relatively easily obtained fuel, with an acceptable cost, which can be distributed by existing service stations with no new infrastructure. However, if this product can be used commercially with no major difficulties, its use to produce electrical energy proves difficult: it is a stable body which liberates ions reluctantly even in the presence of noble catalysts.

The latter must be disregarded due to their rarity. The search for an effective catalyst which is cheap, easily prepared and long-lasting is a big problem. Nor must one underestimate the difficulties relative to technology of the cell itself and its equipment.

Work is progressing, however, and various laboratories have announced positive results with small cells which are still bulky and expensive. However, their great merit lies in showing that this path is open.

Vehicle Characteristics

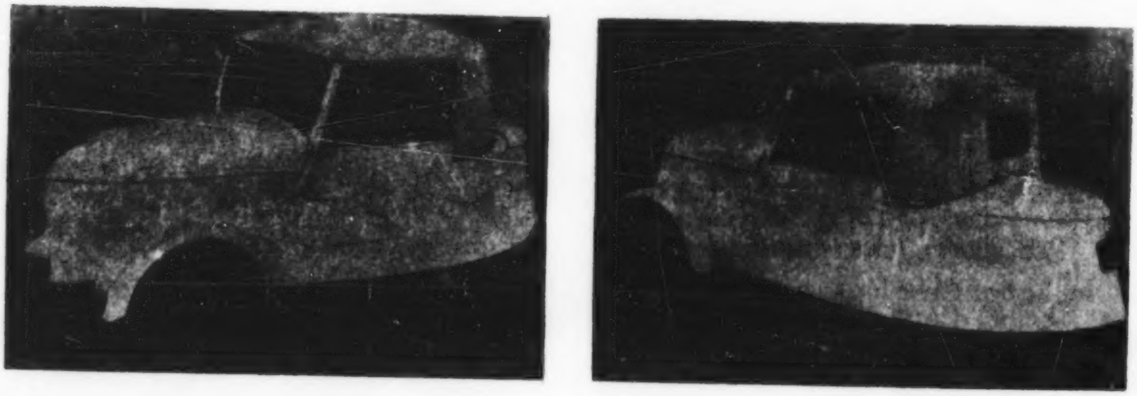
The preceding conditions enable us to define the principal characteristics of the vehicle.

We have seen that, because of safety requirements, it will be very difficult if not impossible to make a very small vehicle. We have also seen that this vehicle will be relatively heavy.

To guarantee adequate commercial success, it will have to have a performance approximating that of comparable traditional vehicles, or at least, not much lower if the users are to be satisfied with it. This point is likewise important for traffic flow since, as we know, traffic flow is better, the more homogeneous it is as far as acceleration and velocity capabilities are concerned.

We can see that the vehicle with a classical lead storage battery is far from meeting these conditions because of its low capabilities. The considerable mass of the required batteries limits the acceleration, and the low mass-energy makes it necessary to limit the velocity. The mass of the batteries itself requires a heavier structure.

The problem of the small vehicle with a storage-battery is well-known by Peugeot, who produced approximately 400 during the years of extreme restriction, 1941-43. It was the VLV, a light, two-seater, town vehicle with lead batteries and, thus, of very limited performance like all vehicles of the type. With this little car, approximately 70 km could be covered with a peak velocity of 36 km/h. Its total weight was 365 kg, of which 160 kg was batteries. Restored and modernized, it is certain that its performance could be increased somewhat. Velocity could be raised to 55 to 60 km/h perhaps, but only by retaining the old type of chassis which does not conform to safety norms.



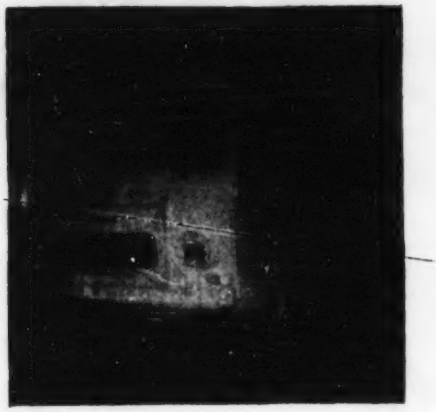
V L V PEUGEOT 1941

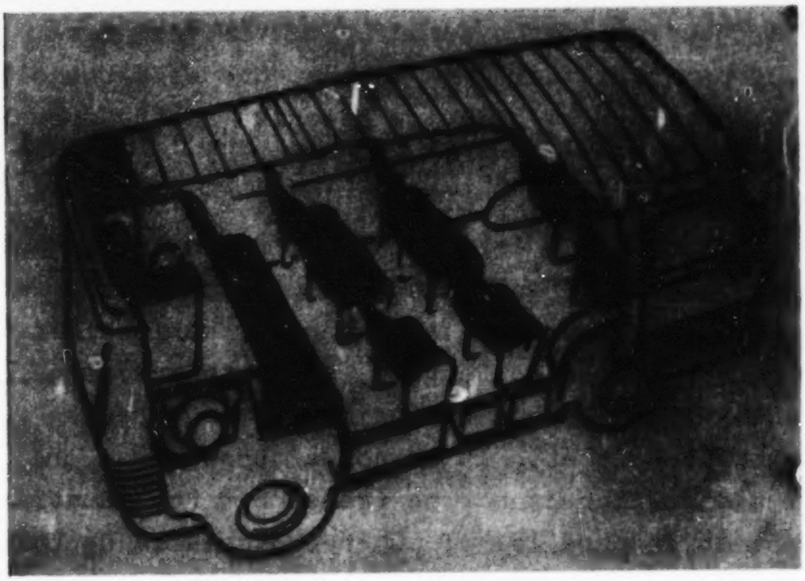
The new batteries being tested will certainly be much better than the present ones, but, short of significant progress, their weight will still limit the possibilities of the vehicles using them.

To obtain the minimal performance defined above, it is necessary to have a generator providing at least 450 wh/kg. Let us recall that lead storage batteries have difficulty in reaching 40 wh/kg. Only a fuel cell appears capable of giving an adequate energy density to ensure velocity, acceleration and action radius.

One should note that the weight of a storage battery is clearly proportional to the stored energy, meaning that the weight increases rapidly with the required action radius. In the case of a fuel cell, this is not the case because the weight of the cell is a function of the power it supplies, and the action radius is a function of the capacity of the fuel tank. With sufficiently powerful fuels, as in the case of methanol, the consumption in kg per kW/h is low relative to the overall weight of the vehicle.

Since the fuel cell is the only solution leading to the production of a vehicle answering the problem posed, we had to find this cell. The studies carried out in this field by the Alsthom Company interested Peugeot, since they were performed in accordance with a method taking into consideration technological problems from the outset. It is not a question of only performing laboratory tests which cannot be transposed to industrial uses. The development in parallel of two techniques is doubtless more difficult, and perhaps slower, but only on the surface because it eliminates the final necessity of spending a great deal of time exploring paths which may not be practical. The two companies collaborated to study a vehicle with a cell. Alsthom, brought to Peugeot its experience in the field of electrical motors and control circuits. This Association has been working since the end of 1967, and interesting results have been obtained.





Above an experimental vehicle is shown. It is a small car capable of transporting 11 passengers plus the driver with 175 kg of luggage. The velocity is 95 km/h.

The vehicle is derived from a type J7 Peugeot, and is to be used for electrical transmission tests.

It is equipped currently as a hybrid with a controlled ignition leading to a constant velocity of the electric generator working in parallel with the batteries. The generator assembly is placed in the rear of the vehicle, and it is anticipated that it can easily be replaced by a fuel cell.

Even in its present form, the performance of the vehicle is comparable to that of traditional vehicles of its category.

In the special field of cells, the level of development obtained with hydrazine cells has interested ESSO-USA, who has just signed an important research contract with the Alsthom-Peugeot Association to develop the methoanol-air cell.

The French Government has recognized the value of the program and is aiding the Alsthom-Peugeot Association by loans supplied through the General Delegation for Scientific and Technical Research.

We have seen the reasons leading Peugeot to develop research on an average-sized electrical vehicle. This type of vehicle satisfies the requirements of reducing nuisances due to atmospheric and noise pollution, and at the same time answers a natural consumer demand.

For commercial success, it is also important that the proposed vehicle need not require too many modifications in the driving habits of the users. They should be able to switch from one vehicle to the other without disorientation or any special constraints. The problem of recharging storage batteries, which requires time and special fixed installations, would always be an unpleasant constraint.

In conclusion, we feel that we should seek to produce a vehicle which, for the user, will differ very little from vehicles currently in use. The electric vehicle with a fuel cell is a solution, but it is still far in the future.

*U.S. GOVERNMENT PRINTING OFFICE: 1972 484-487/323 1-3